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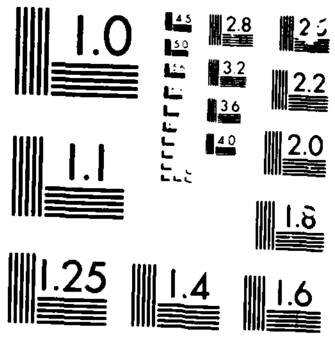

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LITERATURE REVIEW ON THE DESIGN OF COMPOSITE MECHANICALLY FASTENED JOINTS

by

C. Poon

National Aeronautical Establishment

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LITERATURE REVIEW ON THE DESIGN OF COMPOSITE MECHANICALLY FASTENED JOINTS

REVUE DE LA DOCUMENTATION SUR LA CONCEPTION DES JOINTS À LIAISON MÉCANIQUE EN COMPOSITES

by/par

C. Poon

National Aeronautical Establishment

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SUMMARY

This report presents a literature review of the state-of-the-art analytical and experimental methodologies adopted in the aerospace industry for the design of mechanically fastened joints in composite structures. Results and conclusions obtained from the published literature relating to the effects of critical parameters, which include composite material system, fastener configuration and joint geometry, on the mechanical behaviour and failure modes of composite mechanically fastened joints are discussed. Further research required to improve the design of composite mechanically fastened joints is identified as a result of this review.

RÉSUMÉ

Le rapport présente une revue de la documentation sur les méthodes analytiques et expérimentales de pointe, utilisées dans l'industrie aérospatiale pour la conception de joints à liaison mécanique en matériaux composites. Il traite des résultats et des conclusions tirés de la documentation publiée sur l'effet des paramètres critiques, notamment le matériau composite lui-même, la configuration de la fixation et la forme du joint, sur le comportement mécanique et les modes de défaillance des joints à liaison mécanique en composites. La revue indique quelle recherche plus poussée est nécessaire pour améliorer la conception des joints à liaison mécanique en composites.

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LITERATURE REVIEW ON THE DESIGN OF COMPOSITE MECHANICALLY FASTENED JOINTS

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1.0 INTRODUCTION

The purpose of this literature review is to assess the state-of-the-art analytical and experimental methodologies for the design of composite mechanically fastened joints. This review aims at providing a basis for identifying further research in these areas.

Joints that require mechanical fasteners such as bolts, rivets or pins to connect two or more parts in a structure where the transfer of loads is provided by the fasteners are generically described as mechanically fastened joints. This is in contrast to adhesively bonded joints where the connecting and load transfer medium is the adhesive layer. Mechanically fastened joints are required in cases where the need for component disassembly is entailed.

One of the more challenging aspects of composite mechanically fastened joints is that the well-established design procedures for metal joints, that are based on years of experience with isotropic and homogeneous materials, have to be changed in order to accommodate the anisotropic and nonhomogeneous properties of composite materials. Also, advanced composites have practically none of the forgiving capabilities of metals which yield to redistribute loads and thus reduce the sensitivity to local stress concentrations. The inherent matrix weaknesses of composites, especially organic matrix composites, render the joints susceptible to interlaminar shear failures as a result of matrix stresses.

Analytical procedures for the prediction of static strength and fatigue life of composite mechanically fastened joints are presented in Section 2. The application of finite element and two-dimensional elasticity methods in stress analyses and the adoption of failure criteria in static strength predictions are discussed. Current methods for fatigue life prediction are also discussed.

Experiments investigating the effects of important parameters on the mechanical performance of composite mechanically fastened joints are presented in Section 3. Of principal interest in the results discussed are stress concentrations at the fastener hole as a function of fastener configurations and material parameters, and the relationship between failure modes and joint configuration, fastener pattern, lay-up, etc. The special topic of environmental effects is not included in this review.

Further research in improving the design of composite mechanically fastened joint is discussed in the last section. This includes the analytical effort required to improve the accuracy and reliability of both static and fatigue strength prediction methodologies as well as the experimental work required to provide a data base which is essential for the application of advanced high strain/tough resin composites. Also, the development of failure models based on physical damage phenomena is needed for the prediction of delamination and gross bearing failure modes.

2.0 ANALYTICAL METHODOLOGIES FOR STRENGTH PREDICTIONS OF COMPOSITE MECHANICALLY FASTENED JOINTS

A typical analytical procedure for the evaluation of the static strength of composite mechanically fastened joints involves four basic steps: first, the load distribution in the vicinity of the fastener holes is determined by an overall analysis of the structural component; second, the fastener load and the by-pass load at individual fastener holes are determined; third, the detailed stress distribution in the vicinity of an individual fastener hole is evaluated based on the fastener load and by-pass load; and fourth, the joint strength is assessed by applying appropriate material failure criteria. These analytical steps for composite bolted joint strength evaluation are illustrated in Figure 1. Methodologies adopted in each of the steps are discussed in the following sub-sections.

2.1 Overall Structural Analysis

An overall structural analysis to determine the internal load distributions is performed, generally, by finite element methods. Because of economic limitations, it is common practice for a component finite element model to consider overall geometric and material properties to determine stiffness parameters and to exclude fastener flexibility under the assumption that the contributions of bolts and local joint structures to the overall structural deformation are quite small (1). When the bolt flexibility is considered to have an effect on the overall response to loads, the inclusion of fastener effects in the general model is necessary for accurate analysis (2,3). In Reference 2, the finite element analysis of the Space Shuttle payload bay doors clearly demonstrated that the analysis of joint behaviour was required to be an integral part of the overall structural analysis. Also Reference 3 shows that the flexibility of the fasteners was required in the local root area of the overall finite element model of the B-1 horizontal stabilizer. Baumann (4) presented a method incorporating the effects of fastener representation. He discussed various modelling techniques for the fastener effect and demonstrated excellent correlation with test results by allowing the fastener (beam elements) end constraints to be flexible rather than rigidly fixed against rotation.

2.2 Analysis of Load Distribution at Fastener Hole

The overall structural analysis discussed in the previous sub-section can, in most cases, even though fasteners are not modelled, provide an estimation of loads acting on complex joints. One dimensional analytical methods for redundant structures (5) are commonly used to determine the load carried by each row of fasteners in a complex joint. These methods include analytical closed form procedures for simple lap-joint configurations, and numerical procedures capable of handling more complex geometries and joints with multiple shear faces. Engineering idealizations employed in one dimensional analysis are based upon gross assumptions regarding the plate flexibility between successive fastener rows, the bolt flexibility due to shear and bending effects, and the local flexibility associated with the complex stress and deflection pattern in the immediate vicinity of the hole (5,56). As illustrated in Figure 2, rows of fasteners are represented by fastener shear elements in the structural joint idealization.

In order to determine individual fastener loads accurately, it is important to account for the contribution of each fastener to joint flexibility. This contribution is dependent upon fastener stiffness, joint member stiffness, and load eccentricity. Joint flexibilities, which are obtained experimentally from load-deflection tests upon single fastener specimens, are required for the analysis. In metals, this type of data is available for a wide variety of fasteners, sheet materials and thicknesses (6). In composites, however, this data is not as prevalent and is usually generated on a "need" basis for specific conditions. When data is not available, it is common to obtain estimates of composite joint flexibility by comparison to existing isotropic metal data or by calculations using formulae developed for thin sheet metals (3,7).

2.3 Effect of Friction on Bolted Joint Load Distribution

The effect of friction is commonly ignored in the analytical work published in the literature. Friction between plate surfaces can, however, significantly affect joint bolt load distribution. Experimental work by Wittmeyer and Smode (8) and the survey report by Munse (9) both indicate that the clamp-up force resulting from bolt tightening relieves the joint load transmitted by fastener shear. However, in most design situations, this beneficial effect of friction in relieving fastener load is conservatively ignored because it is felt that the bolt torque cannot be maintained due to the viscoelastic property of resin-based laminates which allows bolt clamp-up relaxation (108) to occur during the life of the structure.

In fatigue tests using aluminium single-shear dog-bone specimens with steel Huck rivets, Hooson and Baker (10) reported that failures of specimens occurred not at the fastener hole where the stress concentration is highest, but outside the region of peak clamp-up pressure between plates. Significant fretting was observed in the region of failure. This observation led to the belief that failure was the result of the propagation of cracks which were initiated by a fretting mechanism.

In composites, this contact problem in the faying plate surfaces is further complicated by the fact that the behaviour of friction and wear is a function of varying fiber orientations with respect to the sliding direction. Sung and Suh (11) measured the friction coefficient and wear volume of composites as a function of sliding distance for three different fiber orientations, perpendicular, transverse and longitudinal to the sliding direction (see Figure 3). As illustrated in Figure 4, which presents their results for graphite epoxy composite (Thornel 300/SP-288), both wear and friction coefficients were a minimum when the fiber orientation was normal to the sliding surface, and both wear and friction coefficients were a maximum when the sliding was transverse to the fiber axis. Different failure modes for different fiber orientations with respect to sliding direction were observed in their experiments (Figure 3).

Sandifer (106) investigated the effect of fretting fatigue on graphite/epoxy composites and found that fretting has no significant effect on the fatigue life of graphite/epoxy material when fretted against aluminium, titanium, or graphite/epoxy of the same type. Fatigue life was actually found to be increased by a factor of four under tension-tension cyclic loading due to the clamping of the fretting pad in the test section of the unnotched specimen. It was noted that, during cycling testing, the specimens began to delaminate in the thickness plane between the grips and clamped pads. However, such delamination never occurred in the clamped regions. This observation led to the conclusion that the pads act as a stabilizing point holding the plies together and thus a longer fatigue life is achieved. Sandifer further mentioned that the application of a common test technique where buckling guides or stabilizing fixtures are mounted at specimen mid-point may lead to non-conservative fatigue life results.

The effect of friction in the faying plate surfaces can be included in the stress analysis if the clamping is known. Both finite difference and finite element methods have been applied successfully in calculating the clamp-up pressure for isotropic plates (12, 13, 14). A typical idealization of a bolted joint used to determine the contact pressure between plates is illustrated in Figure 5. The effects of clamping pressure and lateral constraint were investigated experimentally by Stockdale and Matthews (15) on glass/epoxy and by Collings (16) on carbon/epoxy. It was concluded that increasing the bolt torque increases the bearing strength. Semi-empirical equations, which account for friction effects and lateral constraints at the bolt hole, were established by Collings (17) to predict bearing strength and failure mode of carbon fiber-reinforced plastics.

The through-thickness effects for a multi-orientation laminate as a result of fastener/plate interaction are very complicated because the coefficient of friction varies through the thickness, from ply to ply, at the edge of the hole. Also, under compressive and frictional loading, complicated failure modes, such as fibers debonding from matrix and fiber buckling, are encountered. To treat these effects analytically, three dimensional methods and suitable failure criteria are required.

2.4 Detailed Stress Analysis and Static Strength Prediction

The detailed stress or strain distribution in the vicinity of the loaded bolt hole in a composite joint is determined by means of finite element methods, elastic anisotropic analysis based on complex variable formulation and fracture mechanics analysis. The prediction of static strength and failure mode is accomplished by the application of anisotropic material failure criteria based on unidirectional laminate properties.

The failure of a composite laminate is assessed on a ply-by-ply basis. At the edge of the fastener hole where a high stress concentration is present, strength prediction is based on stresses at a "characteristic dimension" from the edge of the hole (18). In this way, the non-linear material behaviour in the region immediately surrounding the fastener hole is avoided. This approach, as illustrated in Figure 6, has been commonly applied in the failure analysis of composite bolted joints (19,20). The establishment of the "characteristic dimension" is based on experimental data obtained by test procedures discussed in References 19-21.

Failure modes in composite bolted joints can be very complex and quite different from those of metal joints because composites exhibit anisotropic properties, lack of ductility and inherent interlaminar weakness. Various failure modes for composite bolted joints are illustrated in Figure 7. Many failure criteria have been developed essentially by modifying isotropic criteria to allow for anisotropic effects in predicting these failure modes and strengths of composite bolted joints. In developing these failure criteria, sufficient arbitrary parameters are introduced so that various failure modes can be incorporated.

2.4.1 Static strength failure criteria

Sandhu (22) published a survey of failure criteria for anisotropic materials in 1972. He broadly categorized these criteria according to their capability to account for failure mode interactions. Failure criteria that do not account for failure mode interactions include maximum stress (23), maximum strain (24), and maximum shear criteria (25). In applying these criteria, failure is precipitated when any one of the longitudinal, transverse, and shear stresses/strains exceed the material limits determined by tests. In the other category of failure criteria where failure mode interactions are accounted for, expressions mainly of a quadratic form that yield a smooth and continuous quadratic failure envelope in each load quadrant, are included. The expressions are either generalizations of Von Mises' criterion, such as those developed by Hill (26), Tsai (27) and Hoffman (28), or have been developed explicitly in quadratic form using the stress tensor approach which satisfies the invariant requirements for coordinate transformation, such as the Tsai-Wu criterion (29). Tennyson (30) adopted the cubic form of the stress tensor polynomial criterion to predict failure strength of graphite/epoxy under biaxial loads and obtained more accurate predictions than with the quadratic form. Experimental procedures required to obtain these parameters for various failure criteria are discussed in Reference 31.

2.4.2 Static strength prediction based on fracture mechanics

Eisenmann (32) established a bolted joint static strength prediction model based on fracture mechanics for composite materials. The failure criterion is:

$$K_I^i = K_Q^i$$

where K_I^i is the Mode I stress intensity factor at location i on the fastener hole boundary and K_Q^i is the corresponding fracture toughness. This fracture mechanics concept is similar to the "characteristic dimension" concept of Whitney and Nuismer (18) except that the characteristic dimension, a^i, is taken as the length of a through crack extending radially outward from location i on the hole boundary. The determination of a^i is based on laminate strength and fracture toughness obtained by tests discussed in Reference 33. Eight potential crack initiation positions on the hole boundary (i=8) are selected based on an examination of many failed joint test specimens. Values of laminate tensile strength and Mode I fracture toughness at these locations are determined by tests using tensile coupons and edge-notched beam specimens fabricated in a manner such that they represent laminate properties in the direction tangential to the hole boundary. Once the laminate tensile strength and Mode I fracture toughness have been determined, the characteristic dimension, a^i, can be calculated for each of the eight selected locations by the following equation:

$$a^i = \frac{1}{\pi}\left(\frac{K_Q{}^i}{\sigma_{ult}{}^i}\right)^2$$

The established dimension, a^i, is then used to calculate the Mode I stress intensity at each of the eight locations and for each of the five specific load cases that consist of the tension loads in the X and Y directions, the bolt bearing loads in the X and Y directions and the shear loads, as illustrated in Figure 8 for location i = 2. The Mode I stress intensity factor for the general load case at location i is obtained by adopting linear superposition of all five Mode I stress intensity factors for specific load cases.

The validity of this fracture mechanics model has been verified by successful correlation of experimental results. Experimental data consisting of measured failure loads and observed failure locations from a series of forty-eight static tensile tests were used (32). The application of this model, however, is limited by the requirement of an extensive data base and is only valid for tensile strength predictions.

2.4.3 Static strength prediction using finite element method

The two dimensional finite element model is by far the most common method in composite mechanically fastened joint analysis (34-49). The major limitation of two dimensional analyses is that three dimensional effects, such as thickness deformation related to bearing failures, interlaminar shear resulting from ply-to-ply displacement incompatibilities, through-the-thickness friction effects between the fastener and the hole, and lateral constraint at the fastener hole as a result of clamping of washer and nut face on the plates that are joined together, are not accounted for. However, in most design situations, two-dimensional methods are chosen over three-dimensional ones because of their relative simplicity and economy.

A two-dimensional finite element method solution predicting bolted joint strength was published by Waszczak and Cruse (34) in 1971. A cosine-distributed radial pressure acting along the semi-circular boundary was used to simulate the load from a rigid and frictionless pin. Orthotropic laminates, which were mid-plane symmetric, were considered. The maximum stress criterion, the maximum strain criterion and the Tsai-

Hill distortional energy failure criterion were applied to predict the laminate failure strength and failure mode. For cases where lay-ups were ±45°, this analysis resulted in failure strength predictions which were 50% conservative.

Chang et al. (35, 36) investigated the same problem using similar techniques. Improved correlations in failure strength and failure mode with experimental results were obtained by adopting the Yamada-Sun shear strength failure criterion (37) in conjunction with a proposed failure hypothesis (36) that predicts failure based on stresses at a characteristic distance from the pin-hole interface in order to minimize three-dimensional effects. Wong and Matthews (38) used the FINEL code to calculate the strains at the pin-loaded holes of mid-plane symmetric and balanced laminates. The layers of a laminate were treated as being homogeneous and orthotropic. One half of the joint was modelled based on symmetry. The load applied by the pin was represented by a sinusoidally distributed pressure as well as by a uniform vertical displacement at the hole boundary on the loaded side of the hole. Only insignificant differences were found between these two techniques of pin load representation. Experimental correlations of results are included in the investigation.

An alternative technique to simulate the frictionless rigid pin joint is to apply zero radial displacements along the semi-circular boundary and to apply force at the far end (39,41,42). Agarwal (39) used this technique and the NASTRAN code to determine the stress distribution around the fastener hole of a double-shear bolt bearing specimen. The composite plate, which was assumed to be orthotropic and mid-plane symmetric, was idealized by 284 CQDMEMI elements which are isoparametric membrane elements and do not include any bending. The plate was assumed to be symmetric about the X axis and only half of the plate was modelled (see Figure 9). The Grimes-Whitney (maximum strain) first ply failure criterion (40) was applied to predict the unnotched laminate strength and the Whitney-Nuismer average stress criterion (19,20) was applied to predict the mechanically fastened joint strength and failure mode. Soni (41) used the same NASTRAN code and the boundary conditions but adopted the Tsai-Wu tensor polynomial failure criterion (29) for the strength analysis of pin-loaded plate. The ultimate laminate failure strength was based on the last ply failure stress. The results obtained by both investigations were conservative by a factor of two for lay-ups which were predominately ±45° when compared with corresponding experimental results.

York et al. (42) used the Structural Analysis Program SAP V and the modified "point stress" failure criterion (43) to predict the net tension strength of composite mechanically fastened joints. Application of the modified "point stress" failure criterion requires the empirical determination of two notch sensitivity parameters, m and c, for a particular material system and laminate configuration. Accurate strength predictions were achieved based on experimental net tension strength data for Hercules AS/3501-6 graphite/epoxy with a laminate configuration of $(45/0/-45/0_2/-45/0/45/0_2/90)_s$.

Crews et al. (44) presented another technique to simulate frictionless pin loading in their two-dimensional finite element analysis where the pin was also modelled. The pin was loaded at its center and was connected to the laminate by short, stiff spring elements which had no transverse stiffness and as a result they transferred only radial loads and thereby produced the desired frictionless interface. An iterative procedure was adopted to determine the contact boundary between the pin and the hole. When a spring was computed to have a tensile force, its radial stiffness was set to zero and the analysis was repeated until convergence was reached. Stress concentration factors, based on nominal bearing stress, for finite size orthotropic laminates of different lay-ups and geometries were established using this analytical technique.

The above methods ignore the effects of friction and the length of contact of the fastener with the boundary of the hole in the laminate. Oplinger (45,46) adopted an accurate treatment of boundary conditions at the fastener hole by modelling fastener/plate interactions in his finite element analysis. This treatment involves the use of a displacement boundary condition to represent the effect of the fastener moving against the hole boundary. The use of displacement conditions in the contact region leads to successful modelling of changes in contact length with increasing by-pass load, a condition which exists in a complex joint with multiple rows of fasteners. The analytical results showing the effect of friction on radial and shear stress distributions around the fastener hole are given in Figure 10. A departure from the commonly assumed half-cosine radial stress distribution as a result of friction is noted in Figure 10.

Wilkinson et al. (47) used an incremental finite element method to determine the stresses and strains around pin-loaded holes in orthotropic plates. The numerical solution provided by the analysis accounts for friction along the contact surface between the pin, which is assumed rigid, and the plate, and determines the region of slip and nonslip. This analytical method was later extended to provide numerical solutions for multiple-bolted joints (48). The effects of variations in friction, material properties, load distribution among the bolts and bolt/plate contact were considered. A condition of nonslip existed at a point on the hole boundary if:

$$\mu\sigma_r \geq \tau_{r\theta}$$

where μ = coefficient of friction, σ_r = radial stress, and $\tau_{r\theta}$ = tangential shear stress. An incremental loading with an iterative procedure was performed to obtain the results at the final specified load level. The effect of friction on the radial stress between the bolt and the contacting hole boundary of a wooden joint obtained from Reference 48 is presented in Figure 11. This figure shows that the total absence of friction ($\mu = 0$) allows the relatively low modulus wood to "wrap" around the rigid pin and thereby distribute the pressure more evenly. For stiffer orthotropic materials, such as glass composite, a change in the contact coefficient of friction from $\mu = 0.7$ to $\mu = 0.4$ has little effect on the radial stress on the boundary of the hole (Figure 12). However, this relative insensitivity of the stress distribution as shown in Figure 12 to moderate changes in friction is fortuitous since, even at a fixed position around a loaded pin, the coefficient of friction for a stacked fiber-reinforced laminate could vary from ply to ply depending on the particular ply's orientation relative to that of the pin in the contact region. The treatment of the through-the-thickness friction effect requires very complicated three-dimensional analysis. No work has been published in this area.

In order to predict bearing and delamination failure and to account for the effect of clamping pressure created by bolt torque in composite mechanically fastened joints, the distribution of stresses around the loaded hole in three directions has to be evaluated. Matthews et al. (49) performed a three-dimensional finite element analysis on a single composite bolted joint by using a new element derived from a standard 20-noded, isoparametric 'brick' element. This modified element can represent several layers of the composite without serious loss of accuracy. The results for three clamping cases were discussed: (1) Pin-loaded hole case where lateral constraint is excluded; (2) finger-tight washer case where lateral constraint is provided; and (3) bolted joint case where a compressive displacement to all the surface nodes under the washer is imposed. The effect of friction was ignored in the analysis. It was observed that when the laminate is loaded via a bolt with finger-tight washers, the most noticeable change from the pin-loaded case is a reduction of the through-the-thickness tensile stress. This observation was consistent with the increase in failure load obtained experimentally. For the bolt loading with a fully clamped washer, a significant increase in the direct stress, σ_{ZZ}, under the washer and the interlaminar shear stress, σ_{ZX}, at the edge of the washer in the outer plies, was noticed. Again this is consistent with experimental results where failure was found to occur by delamination at the washer edge. A suitable failure criterion has not been combined with the stress analysis to predict the actual failure loads and failure modes.

2.4.4 Static strength prediction using elastic anisotropic analysis

These methods are principally formulated from two-dimensional anisotropic elasticity theory (50). In these methods, the stress distributions around a hole in an infinite orthotropic laminate are determined and various ways of correcting these stresses for finite laminate widths and lengths have been applied (51,52). There are two common techniques of modelling fastener radial load distributions: (1) a radial stress boundary condition varying in a cosine distribution (52,53,55); and (2) a radial displacement boundary condition corresponding to rigid displacements of the fastener coupled with a solution of the associated contact problem (58,59). In most cases, fastener frictional shear forces at the hole boundary have been ignored.

Waszczak and Cruse (53) solved the problem of an infinite anisotropic plate containing a circular cut-out. The plate was loaded by the bolt load, which was represented by a cosine distribution of normal stress and was subjected to a uniform stress field caused by tension loads applied at two far ends of the plate.

The method of superposition was used to generate the solution to the problem of interest by combining two infinite plate solutions. One case contained the bolt loading only while the other case contained the tension loading only. A series solution based on the theory of anisotropic elasticity was derived for the bolt loading case. The solution to the case of a plate with a hole under tension loading was obtained from Reference 54. Both infinite plate solutions for the two cases were corrected for the effects of finite specimen size using anisotropic correction factors *generated by* Boundary Integral Equation methods (51) prior to their superposition. Pin/plate interaction was assumed to be frictionless. It was noted that the use of correction factors to modify the infinite plate solutions produced a stress field which no longer strictly satisfies overall equilibrium requirements.

The maximum stress, the maximum strain and the Tsai-Hill criteria were considered for static joint strength predictions based on a first ply failure hypothesis. Conservative predictions of failure loads were obtained. The degree of conservatism was found to be a function of specimen lay-ups varying from 2% for a $(0_6/\pm45°_5)$ boron-epoxy laminate to 53% for a (±45°) boron-epoxy laminate where large shear deformation occurred. Prediction of failure locations was found to be satisfactory.

De Jong (52) presented a solution of the stress distribution around a pin loaded hole in an orthotropic plate. The approach used by De Jong was similar to that used by Waszczak and Cruse (53) except that the normal stresses carrying over the fastener loading force on the boundary of the hole were represented by a sine series where the coefficients of this series were calculated from the boundary conditions for the displacements of the loaded section at the edge of the hole. Waszczak and Cruse (53) only used the first term of the cosine series as a stress boundary condition and the possibility of determining the normal edge stresses in relation to material properties by means of a displacement boundary condition was not exploited. A superposition technique was then adopted to estimate the stresses in the plates of finite widths from infinite plate results. The prediction of joint strength by failure criteria was not investigated. One of the conclusions reached by De Jong was that although the pin has a neat fit in the hole, there is a clearance, resulting from elastic deformations of the plate material, not only between the pin and the unloaded side of the hole, but also between the pin and a small region of the loaded side as well.

Garbo and Ogonowski (55,56) developed a Bolted Joint Stress Field Model (BJSJM) which utilizes two-dimensional elastic anisotropic theory to determine laminate stress distributions around an unloaded or loaded fastener hole in orthotropic materials. The principle of elastic superposition was used to obtain laminate stress distributions due to the combined bearing and by-pass loading. Loaded hole analysis was performed by specifying a radial stress boundary condition varying as a cosine function over half of the hole. The stress solutions obtained are valid for mid-plane symmetric laminates only. Strain distributions are calculated using material compliance constitutive relations. Laminate compliance coefficients were derived from classical lamination plate theory (57) with unidirectional material elastic constants, ply angular orientations, and ply thicknesses. Strains for individual plies along lamina principle material axes were calculated using coordinate transformations. Finite width effects were accounted for by the superposition technique adopted by De Jong (52). To minimize the effect of nonlinear material behaviour at the hole boundary, the "characteristic dimension" hypothesis of Whitney and Nuismer (18) has been adopted in BJSFM. Laminate failure was predicted by comparing elastic stress distributions with material failure criteria on a ply-by-ply basis. Various material failure criteria, such as Tsai-Hill (27), Hoffman (28), Tsai-Wu (29), maximum stress (23), and maximum strain (24), were incorporated in the BJSFM. Analytical predictions of joint strengths provided by the BJSFM have been extensively calibrated against experimental results (55,56).

Oplinger and Gandhi (58) presented the results of stress distributions around a hole in a pin loaded orthotropic plate by using a two-dimensional anisotropic elastic analysis which employed a least-squares boundary collocation scheme. The mode of interaction between the pin and the plate was described by a radial displacement boundary condition corresponding to a rigid displacement of the pin in the region of contact together with a condition of zero radial pressure outside the contact region. Iteration techniques were used to solve the non-linear contact boundary conditions of this problem. In a related investigation by Oplinger and Gandhi (59), results are presented which describe the effect of Coulomb friction between the pin and plate on the radial and shear stress distributions around fastener hole. These results are illustrated in Figure 10 for friction coefficients ranging from 0 to 0.5. Significant effects of friction on stress distributions are displayed in Figure 10.

2.5 Fatigue Life Prediction Methodology

There are basically four methodologies adopted for predicting composite fatigue behaviour. These methodologies are: (1) empirical correlation, (2) cumulative damage model, (3) residual strength degradation model, and (4) tensor polynomial failure criterion. The empirical approach has been extensively applied in fatigue life prediction of mechanically fastened joints in composite structures. Only limited experimental verifications of the accuracy of fatigue life prediction for composite mechanically fastened joints have been carried out for the remaining three methodologies. A review of the four fatigue life prediction methodologies is presented in the following:

(1) Empirical methods - current state-of-the-art fatigue verification approaches for composite structures employ spectrum fatigue tests on components/specimens representative of specific design details (60). These empirical methods are extensively applied due to a lack of confidence in existing analytical composite fatigue life prediction procedures which still require more experimental calibration. In all modern military aircraft that contain extensive composite contents in their primary and secondary structural components (e.g. B-1, F-15, F-16, F-18, AV-8B), empirical methods have been used extensively to assess the effects of cyclic loading on composite fatigue life in order to comply with various military durability specifications such as MIL-A-8866, MIL-A-83444 and MIL-STD-1530A (61-66). It has been postulated that sufficient fatigue life can be achieved by composite structures designed to satisfy static strength requirements (63). For the composite wings of the F-18 and the advanced Harrier aircraft (AV-8B), the maximum design strain level has been limited by McDonnell Douglas Aircraft Company in the range of 4000 to 5000 μ inches/inch. These design strain levels have been developed to accommodate the stress concentration effects of fastener holes and also serve to provide an inherent damage tolerant structure (63).

Most research and development programs on composite fatigue have also emphasized experimental investigations. Conclusions and recommendations reached in these studies have been based on empirical curves fitted through data. Generally, physical understanding of the failure mechanism involved is not included. This makes the extrapolation of curves very difficult or even meaningless. Most published fatigue data have been on unnotched laminates or laminates with an unloaded hole. Relatively little fatigue data exist on composite mechanically fastened joints. Of the existing data, results are often for specialized specimen design, lay-up, or test conditions (67,68).

(2) Cumulative damage model - Miner's linear cumulative damage rule is the most commonly applied cumulative method for analyzing composites because of its relative simplicity. This method requires only constant amplitude fatigue data (S-N curves) for the applied stress ratios in the spectrum in order to predict fatigue life. A simplistic spectrum fatigue life prediction procedure for composites using Miner's rule is illustrated in Figure 13.

There are disagreements reported in the literature regarding the accuracy of the composite fatigue life prediction using Miner's rule. In some cases, it has been reported that Miner's rule is grossly unconservative in predicting life of composite materials (69,70). Others have found it to be an adequate technique for preliminary design studies (71). An investigation at McDonnell Douglas Aircraft Company (1) has found that Miner's rule is adequate to gauge the severity of spectra variations.

The large amount of scatter in composite fatigue life may be the main reason for unreliable analytical predictions that have led to disputable conclusions. One of the major difficulties in developing a composite fatigue life prediction method is to provide sufficient replicate testing in order to establish statistical scatter factors to account for the variability of composite fatigue life.

(3) Residual strength degradation model - Yang (72) derived a residual strength degradation model to predict the fatigue life of composites. This model was derived based on the assumption that residual strength is a monotonically decreasing function of the applied load cycles. Weibull statistical procedures are used in this model to predict residual strength and fatigue life. Parameters required in the analysis are derived from static and constant amplitude fatigue (S-N curves) test data. Once these parameters are derived, probability of survival curves can be generated. The major limitation of this approach is in the basic assumption of continuously decreasing residual strength which makes the model incapable of accounting for initial strength increases that have been observed in many investigations on fatigue of composites (73).

(4) Tensor polynomial failure criterion - Tennyson et al. (30) have extended the application of the tensor polynomial failure criterion from static strength prediction to the fatigue life prediction of composite laminates. Unlike static strength parameters, the fatigue strength parameters are not constants, but rather are functions of the frequency of loading (n), the number of cycles (N) and the stress ratio $R = \sigma_{min}/\sigma_{max}$, i.e. $F = F(n,N,R)$. "Fatigue functions" required to predict the fatigue life of a laminate under uniaxial tension and compression cyclic loading conditions with constant frequency and R

ratio have been established for the plane stress condition and implemented into a quadratic formulation of the tensor polynomial failure criterion (30). These fatigue functions were established based on results from tension and compression tests in both the fiber (1) and transverse (2) directions, as well as pure shear in the 1-2 plane. Applications of this model to predict fatigue life of "flawed" and "unflawed" graphite/epoxy laminates for uniaxial load cases in hot/wet environments with thermal-spike cycles were attempted and some encouraging results were reported (30). Current work involves applying this model to predict the fatigue life of graphite/epoxy laminates under random FALSTAFF loading conditions. Some experimental data using a four point bending specimen have been generated.

Rotem (74) has established a general fatigue failure criterion also based on experimentally determined "fatigue functions" for multidirectional laminates. "Fatigue functions" that account for delamination have also been developed. The effect of temperature is accounted for by experimentally determined "shifting factors" for the "fatigue functions" (75).

Presently, it is uncertain which methodologies provide the most reliable predictions, and under what conditions they are reliable. Thus it is important for the reliability aspects of the fatigue prediction methodologies discussed earlier to be evaluated. Until this is complete, it remains a difficult task for designers or researchers to select a methodology than can provide reliable fatigue life predictions under their specific requirements and conditions of interest.

3.0 EXPERIMENTAL INVESTIGATIONS ON COMPOSITE MECHANICALLY FASTENED JOINTS

In the process of preparing this review, it was observed that the majority of the published work on composite mechanically fastened joints included comparisons of experimental results. This is mainly due to the fact that experimental investigations are often required to characterize the complicated behaviour of composite mechanically fastened joints which cannot be treated solely by the analytical methodologies described in the previous section.

State-of-the-art empirical approaches in composite joint strength analysis represent an alternative, often regarded as an expensive one, to the detailed stress distribution analysis described in the previous section. Through tests on design oriented composite specimens, the failure strength of a specific mechanically fastened joint as influenced by parameters such as geometry, lay-up, percent of load transferred in joint through bearing and by-pass etc., is assessed. However, it is obvious, in view of the very large number of variables involved, and their effect on each other, that a complete characterization of a general joint behaviour is impractical. Rather, the current approach is to determine as thoroughly as possible the behaviour of a few basic joints in a limited number of material systems and to hopefully infer the influence of the more important parameters from which the behaviour of other joints and materials can be predicted by empirical design methods. Hart-Smith's work is an example of a comprehensive experimental investigation of some mechanically fastened joints in graphite/epoxy composites where experimental results were generated to establish empirical formulae and design-analysis procedures (76).

Experimental results are also generated in order to complement/verify analytical results obtained by methodologies described in the previous section. The advantage of this is that once an analytical methodology has been well calibrated against experimental results, it can be used to predict composite mechanically fastened joint strengths and thus reduce the high cost of experimental assessment in new design applications.

In the following sub-sections, experimental results and conclusions obtained from published literature relating to the effects of various parameters on composite mechanically fastened joints are presented and discussed. For convenience, the parameters are arbitrarily divided into three groups:

(1) Material parameters: fiber type and form (unidirectional, woven fabric etc.), resin type, fiber orientation and stacking sequence.

(2) Fastener parameters: fastener type, fastener size, clamping force, washer size, hole size, and tolerance.

(3) Design parameters: joint type, laminate thickness, geometry (pitch, edge distance, hole pattern, etc.) load direction, load mode (static or cyclic), and failure definition.

3.1 Material Parameters

Fibrous composites, in most secondary and primary aeronautical applications, are generally manufactured by stacking layers of prepreg consisting of reinforcing fibers embedded in a resin matrix. Graphite, Kevlar or glass fibers and epoxy resins are common ingredients used in producing laminates. The reinforcing fibers are arranged in either unidirectional or woven format in a single ply of prepreg where they are saturated with resin material. This resin matrix serves to bind the fibers together and transfer loads to the fibers.

The mechanical behaviour and failure mode of composite mechanically fastened joints are dependent upon the orientation and stacking sequence of plies in the laminates. In their work on unloaded holes in laminates, Rybicki and Schmuerer (79), and Pagano and

Pipes (80) demonstrated that the stacking sequence of plies affects the interlaminar normal and shear stresses around the unloaded hole and hence, by inference, the strength of a loaded hole in a composite mechanically fastened joint. In order to reduce these matrix stresses which are responsible for delamination at the fastener hole or other free edges, it is important to intersperse the ply orientations thoroughly in the laminates such that the number of parallel adjacent plies are minimized (76).

The effect of stacking sequence on the bearing strength of composite bolted joints was investigated experimentally by Garbo and Ogonowski (56) and Ramkumar and Tossavainen (77) by grouping plies with the same fiber orientation together in the laminates. Both groups of investigators found that the bearing strength decreased when the percentage of the parallel adjacent plies with the same fiber orientation was increased. Quinn and Matthews (78) investigated the effect of stacking sequence in glass fiber-reinforced plastics. They showed that placing 90° plies perpendicular to the load direction at or near the surface improved the pin bearing strength.

The effect of orientation of plies or lay-up on the bearing strength of composite mechanically fastened joints has been investigated by several authors. Collins (81), whose work covers bolted joints in graphite/epoxy composites, concluded that for optimum bearing properties, more than 55% but less than 80% of 0° plies (i.e. parallel to the load) are required, the balance being made up of ±45° plies to provide transverse integrity to the composite bolted joint. He also concluded that optimum tensile properties were obtained when the ratio of 0° to 45° plies was 2:1 whilst optimum shear strengths required a ratio of 1:1. Matthews et al. (83) performed tests on composite riveted joints and concluded that the bearing strength is significantly higher for the 0°/±45° lay-up than for the 90°/±45° lay-up.

Ramkumar and Tossavainen (77) investigated the effect of lay-up on the strength of laminates that were bolted to metallic plates using a single fastener. They tested laminates that were fabricated using nonwoven AS1/3501-6 graphite/epoxy material with lay-ups that ranged from a fiber-dominated lay-up to a matrix-dominated lay-up. They found that, under compression loading, the failure strain increased with an increase in the percentage of ±45° plies while the gross compressive strength and bearing strength decreased with an increase of the percentage ±45° plies. Their compression test results and tension test results are presented in Figure 14 and 15 respectively. They observed that the failure mode changed from the shear-out mode to local bearing failure mode when the percentage of ±45° plies was increased from 40% to 60% under tension loading. However, under compression loading, they found that failure mode was insensitive to lay-up and specimens tested always failed in the local bearing mode.

The mode of failure is also influenced by joint geometry. This aspect will be discussed in sub-section 3.3.

It is recognized that a high stress concentration factor exists at the fastener hole of a composite mechanically fastened joint. Berg (83) emphasized that because most composite materials are incapable of yielding, local load redistribution at the fastener hole can only be achieved by fiber fracture. The distribution of stresses at the fastener hole is influenced by the lay-up of the laminates. Collings (84) suggested the inclusion of $\pm 45°$ plies in order to reduce the stress concentration at the fastener hole.

Hybrid laminates can also be used to reduce the sensitivity to stress concentration. Hart-Smith (76) replaced some of the graphite fibers which were aligned with the load direction with S-glass fibers. Mechanically fastened joint specimens fabricated from these glass/graphite hybrid laminates were consistently as strong or stronger than the equivalent all-graphite specimens when tested under tension loading. However, because of a lower modulus for the glass fibers with respect to the graphite fibers, the stabilization of compressively loaded joint specimens was found to be a problem. The failure mode of the glass/graphite was almost exclusively associated with local bearing failures rather than the potentially catastrophic tension-through-the-hole failure which was common for many of the all-graphite specimens.

The experimental results discussed here are essentially based on conventional material systems composed of graphite fibers of moderate modulus and first generation brittle resin materials (The classification of resin materials is according to Johnston (85)). Advanced composite material systems composed of high strain graphite fibers and second generation tough resin materials are now being manufactured by the composites industry. A survey was conducted by Canadair Ltd. (86) on the types of advanced material systems which are being examined by the aeronautical industry for the next generation of aircraft. As indicated by the survey report of Canadair Ltd., the tensile strengths and the compressive strengths of the advanced composite materials are significantly higher, by as much as 87% and 22% respectively, relative to those of the conventional baseline systems. The most important improvement of the newer composites over the conventional composites appears to be the post-impact performance. Unfortunately, there is no mechanically fastened joint data available currently in the published literature for these newer material systems.

3.2 Fastener Parameters

Many types of fastener are used in aerospace manufacturing. Some common types are screws, rivets and bolts. Each type of fasteners can be offered in a wide variety of dimensions, configurations and materials. The selection of fasteners depends on the type of applications. In general, screws give the lowest load-carrying capacity and tend to be of little use in a primary structural role. Both rivets and bolts offer adequate strength in composite joints for medium to high load transfer applications.

In composite structure, some parameters which affect the selection of fasteners are edge and side distances, hole diameter, laminate thickness, fiber orientation, laminate stacking sequence, and the type of material systems being used. For example, composite laminate thickness, material and location in an airframe structure are factors to be considered in determining whether a blind or two-piece fastener is selected. If the material stack-up has a thin top sheet, the hole countersink configuration and the head configuration of the fastener become important considerations.

Composite materials pose special problems for mechanical fastening because of their peculiar properties. In their survey report on fasteners for composite structures (88), Cole et al. identified four primary problems: (1) galvanic corrosion; (2) galling; (3) installation damage; and (4) low pull-through strength. The nature of these problems and the design of fasteners to circumvent them are discussed below:

(1) Galvanic corrosion: the basic force of the galvanic corrosion reaction is the difference in electrode potential between the graphite fibers and the metals. The less noble metals may corrode when mechanically fastened to graphite fiber composites. One solution is to cover the fastener with a protective coating. Prince (87) performed a comparison of the effectiveness of different coatings to protect against galvanic corrosion. He concluded that, when flawed, coatings are inadequate to provide protection against corrosion.

A more effective solution to the corrosion problem is to select compatible materials for the fasteners. For this purpose, a galvanic compatibility chart, shown in Figure 16, is used. This chart ranks nobility, or resistance to galvanic corrosion of fastener metals in a graphite based composite. Titanium is one of the most noble metals, and so titanium and titanium alloys offer excellent corrosion resistance when used with graphite composite.

A special composite fastener, made of both graphite/polyimide and glass/epoxy composites, has been developed to provide total compatibility, low weight, and low cost (88). The most serious disadvantage of this type of fastener is the lack of reliability of the adhesive bond which holds the two-piece fastener together.

(2) Galling - Galling problems are encountered when nuts fabricated with either titanium or A286 CRES steel are used with titanium bolts, such as the common Hi-Lok system (88). A lock-up situation occurs during installation prior to the development of the desired preload. McDonnell-Douglas has successfully eliminated this problem in the F-18

and AV-8B programs by applying suitable lubricants (88). Galling problems were also encountered in the Lockheed L1011 Advanced Composite Vertical Fin program. The solution was to use stainless steel nuts (Type 303) to replace A286 CRES steel or titanium nuts (88). Other solutions to the galling problem include the use of free running nuts, as in the Eddie Bolt system, and swaged collar fasteners such as the Groove Proportional Lockbolt (GPL) made by Huck Manufacturing Company.

(3) Installation damage: the procedure for installation of fasteners in metallic structures often uses high preload and interference-fit to obtain strength and durability improvements in the joint. Experience has shown that using the same fasteners and procedures in composite structures can produce unacceptable damage. Before the installation of fasteners, holes must be cleanly drilled through the composite structure. This is not an easy task because composites are prone to fraying within the hole and to splintering on the exit side of the hole as a result of the drilling operation; consequently, special drill bits and procedure different from those used in metal drilling, are required in order to avoid introducing any local weakening of the composite structure. While the fastener is being installed into the hole, its rotation is not recommended because this would lead to breaking and lifting of fibers from the surface. Also, axial misalignment of fastener in the hole could lead to damage. For example, misalignment of a blind fastener during installation could cause the blind head to dig into the surface of the hidden side, crushing or delaminating the composite skin.

Several manufacturers have introduced fastener designs which have significantly facilitated their installation in composite structures. Figure 17 shows a typical design approach and installation sequence of a blind fastener. Normally, hydraulic or pneumatic tools are used to install the fastener after it has been inserted into a prepared hole. An axial pulling force is applied to cause a sleeve to form an expanded head on the blind side. This head expansion is accomplished by buckling the sleeve either against the composite structure or against a shoulder on the fastener shank. The fastener is designed to ensure hole fill-in and axial alignment during installation. Mechanical locking is achieved when the head is secured against the blind-side surface. A break groove is normally designed into the stem so that it fractures after a certain level of clamping force has been reached.

Figure 18 illustrates some unique features of a special blind fastener designed to overcome the problem of crushing the surface near the edge of the fastener hole as a result of buckling the sleeve directly against the composite structure. It has a washer element between the corebolt and the sleeve, and it is independent of both. The washer is driven over the tapered end of the nut and it expands to its final diameter. The formed washer is then seated against the joint surface by the continued advance of the sleeve and the corebolt. The unique feature of this fastener is that its blind head is formed by expanding its washer element to the desired diameter before engaging the sheet surface; consequently, there is no surface pressure brought to bear on the composite structure to cause any crushing damage. A straight axial load is applied to deliver the required clamping force, so there is no spinning action between fastener and structure which could cause fibers to separate.

Figure 19 illustrates the installation sequence of a two-piece lockbolt fastener for solid composite laminates. The fastening principle is based on the straight line tension/tension or swaged collar concept which minimizes the tendency of crushing the surface of the composite laminates. Furthermore, this swaging action fills the annular locking grooves on the pin with the collar material to form a permanent lock; thus the rotation of the nut is eliminated. This type of lockbolt fastener is designed for installation in clearance or interference fit conditions.

The nut tightening, collar swaging or tail forming during the installation of a fastener exerts a clamping force or preload on the composite sheets being joined. Because of the low through-the-thickness shear and compressive properties of composites, high preload may result in composite crushing. In order to achieve higher preloads to improve joint performance while minimizing the possibility of crushing damage, fasteners are designed with enlarged "footprints," which refer to the bearing area of the nut, collar or tail, and enlarged heads (Figure 20). Both serve to provide a larger area over which the preload can be spread. This increases the performance of the joint by allowing higher preloads to be achieved which lead to improved shear strengths to resist fastener cocking associated with eccentric loads and improved tensile strengths to prevent fastener pull-through. In addition, higher preloads increase the fatigue life of composite mechanically fastened joints (56).

The low interlaminar strength of composite materials often leads to delamination of the plies on the backside of the laminate when fasteners are forced into an interference-fit hole. Also, experience with fiberglass has shown that when rivets are installed to completely fill the hole by shank expansion, ply delamination and buckling occur at the hole boundary. These difficulties have led to the general requirement for all material systems that only clearance fit (-0.000 to +0.004 inch) fasteners are used in composite structures (88). However, interference fit is desirable in composite joints. Sendeckyi and Richardson (89) demonstrated that increasing the level of interference in a graphite/epoxy joint increases the fatigue life. Interference is also required to increase the load-sharing capability in a joint with multiple rows of fasteners, to prevent fuel leakage in fuel tank area, and to provide the reactive torque needed for one-sided installations of blind fasteners.

Several manufacturers have produced special fasteners for interference installations. Grumman (88) developed the stress-wave rivet system for installation of titanium and A-286 CRES steel rivets in composite structure with up to .008 inch interference. The rivet deformation in the system is caused by a stress wave which causes the material to flow outward in all directions simultaneously. The damage is limited by this simultaneous tail formation and hole filling action. Huck Manufacturing Company has developed the HUCK-TITE titanium interference fit lockbolt. This fastener can be installed conveniently with standard installation tooling developed by Huck. The installation procedure is illustrated in Figure 19. The sleeve of this fastener is expanded into an interference fit hole during installation; consequently, an intimate contact is established between the fastener and the hole which forms a water intrusion barrier to prevent fuel leakage without the application of a sealant and also provides electrical continuity to prevent arcing.

(4) Low pull-through strength: the pull-through strength of a fastener for composite structures is shown to be a function of the size of the "footprint" and the outside diameter of the head as illustrated in Figure 20. An improvement in the pull-through strength can be achieved by using fasteners with enlarged "footprints" and enlarged heads. In the case of blind fasteners, many engineers feel that the blind head should be expanded 1.4 times the shank diameter instead of the 1.2 times used for typical metal fastening (108). The configuration of the head also has significant effect on the pull-through strength. Boeing has found that the 130° shear head can support 30° more load than the 100° shear head because of the improved bearing surface area (88). Lockheed-Georgia Company compared the performance of four head configurations (100° shear, 100° tension, 120° shear and 130° shear) on the basis of pull-through strength and fatigue. The results of the comparison indicated that, for sufficiently thick composite structure where the thickness is in excess of the fastener head height, the 100° tension flush head fastener is the best among the four configurations tested (88).

3.3 Design Parameters

In the primary and secondary structures of aircraft components that utilize composite materials, mechanically fastened joints of various composite-to-composite or composite-to-metal designs can be found (90, 91). Although these joints may be complex in appearance, each can be generically modelled as simple single or double lap specimen.

In selecting the type and configuration of a joint specimen, the following design parameters that reflect the structural joint requirements must be considered: (1)

geometry, (2) hole pattern, (3) hole size, (4) laminate thickness, (5) load eccentricity, (6) fastener load direction, and (7) failure definition. The results of the survey with respect to these design parameters are presented in the following sub-sections.

3.3.1 Geometrical Effects

The convention adopted in this review, which is similar to that suggested as a standard by Kutscha and Hofer (92), is shown in Figure 21.

The effect of end distance (e) has been investigated by a number of authors and they all agree that a certain minimum value of the e/d ratio is required to develop full bearing strength. Ramkumar and Tossavainen (77) investigated the strength of bolted laminates using ASI/3501-6 graphite/epoxy unidirectional prepreg material. They concluded that the bearing strength of all the lay-ups increases with e/d ratio to a value of 4 or 5, beyond which the bearing strength is relatively invariant. The 50/40/10 lay-up, where the number indicates the percentage of the 0°, $\pm 45°$ and 90° fiber orientation in the laminate respectively, exhibits a shear-out mode of failure when $e/d \leq 3$. For $e/d > 3$, the laminate exhibits a local bearing mode of failure. The 70/20/10 lay-up exhibits a total shear-out mode of failure for e/d values below 5. Collings (16) in his work on bolted joints in CFRP showed that for an all $\pm 45°$ lay-up, the minimum e/d ratio required for the development of full bearing strength in the laminate is 5, whereas for a pseudo-isotropic lay-up, the minimum e/d ratio is 3. MIL-HDBK 17A (93) quotes a minimum e/d ratio of 4.5 for 0/45° lay-up.

The width effects in single hole joint specimen have also been investigated in conjunction with the end effects. The results obtained from single hole joints are often used to predict the pitch distance effects in multi-hole arrangements. This is done by representing individual bolts isolated from a single row in a strip of a width equal to the bolt pitch. Ramkumar and Tossavainen (77) showed that for $w/d > 6$, the bearing strength remains relatively constant. When $w/d < 4$, the failure mode is primarily a net section failure across the hole. For $w/d > 4$, the failure mode is primarily a partial or a total shear-out of the 50/40/10 laminate. For the same w/d ratio, the 30/60/10 laminate fails in a local bearing mode. Collings (16) suggested minimum w/d of 8 and 5 for $\pm 45°$ and pseudo-isotropic lay-ups respectively, if full bearing strength is to be developed. Matthews et al (82) showed that for $0/\pm 45°$ lay-ups in CFRP a minimum w/d of 4 is needed.

3.3.2 Hole Pattern

The preceding sections have dealt with single-bolt joints where failure can be defined uniquely in terms of bolt load alone. In most practical applications, however, this is not the case because the load is frequently transferred in multi-row fastener patterns, such as at a chordwise splice in a wing skin, or along a bolt seam aligned with the dominant load path, such as at a wing spar cap. In such more complex load situations, it is necessary to characterize both the bolt load and also the general stress field in which the particular bolt under consideration is located.

Geometrical parameters can influence the amounts of load transfer, as depicted in Figure 22 (110). The secondary bending is created by the load transfer in single shear or otherwise excentric joints even if the external load is free from bending moment. Also, it is affected by geometrical changes. For a single shear joint with two rows of fasteners, it has been demonstrated that by increasing the distance between the rows from 1.57 inches to 3.15 inches, the secondary bending was reduced by 35% (110).

Agarwal (94) investigated the behaviour of multi-fastener bolted joints in AS/3501-5 graphite/epoxy. His experimental results indicated that the net tension failure stress of the joint is increased slightly (5 to 10 percent) by increasing the number of fastener rows. It was also noted that the net tension failure stress was reduced by up to 15 percent as the number of fasteners in a row was increased which was suggested to be caused by the specimen width effect. Hart-Smith (76), using T300/5208 graphite/epoxy, also demonstrated that for joint geometries producing tension failures for a single bolt, the addition of further rows of bolts generally increases the joint strength very little. He found that only when bearing failures occur do multi-row bolt patterns increase the joint strength significantly above the strength of a single bolt row. He observed that the transition between tension and bearing failure modes occurs in the range of bolt pitches between four and six times the diameter of the bolt hole.

Godwin et al (95) presented the experimental results of multi-bolt joints in glass-reinforced plastics. It was shown that bolts in a row develop full bearing strength at pitches of more than six diameters. At small values of pitch in a wide panel, it was suggested that the joint strength could be increased by increasing the end distance to suppress shear-out failure. It was also observed that no substantial improvement in strength is gained by using staggered rows of bolts. They suggested that the joint geometry that optimizes the joint strength is a single row of bolts, at a pitch of 2.5 diameters and an end distance of 5 diameters. At these values, joint strength is approximately half the gross panel strength, and the failure mode is in tension at the minimum section. They pointed out that increased safety can be gained by increasing the pitch to 5 or 6 diameters if there is bearing failure. In this case, the net joint strength is only about one-third of the gross panel strength. This value of pitch compares closely with that suggested by Hart-Smith for graphite/epoxy.

3.3.3 Hole Size

Garbo and Ogonowski (56) investigated the effects of hole size on composite bolted joint strength by applying tension to the two-fastener-in-tandem double shear specimens and loading to failure. It was shown that both bearing strength and gross joint failure strain decrease with increasing hole diameter over the range tested. It was observed that all specimens failed in bearing-shearout, regardless of hole size. Ramkumar and Tossavainen (77) investigated the effect of fastener diameter on the tensile response of a 20 ply, 50/40/10 lay-up graphite/epoxy laminate in a single lap configuration. It was shown that the gross tensile strength and the bearing strength of the laminate decrease when the fastener diameter is increased. A similar trend is also observed under static compression.

3.3.4 Laminate Thickness

Garbo and Ogonowski (56) also investigated the effects of laminate thickness and fastener countersink on graphite/epoxy laminate bearing strength. Their investigation covered a range of laminate thicknesses from 20 ply to 60 ply and three countersink depth-to-laminate thickness ratios (0.77, 0.38 and 0.26). Their test specimens were of a single fastener in double shear configuration with 50/40/10 lay-up and were loaded to failure in both tension and compression. The bearing strength was shown to increase with increasing laminate thickness from 20 ply to 60 ply. Within data scatter, it was observed that the effects of countersink versus noncountersink on strength appear to be insignificant. For pin-loaded holes, i.e. no through-thickness clamping, Collings (17) showed that bearing strength reduces as d/t increases. Ramkumar and Tossavainen (77) showed that an increase in the thickness of a laminate in a single lap configuration introduces additional load eccentricity and bolt flexibility effects. Their experimental results from tests on 20-ply and 60-ply laminates with 50/40/10 lay-ups demonstrated that the thicker laminate strength was approximately 5 percent lower than that of the 20-ply laminates.

3.3.5 Load Eccentricity

In selecting a single lap joint specimen for an experimental program, joint eccentricity effects must be minimized or accounted for because significant bolt bending can lead to a lower joint strength due to the eccentricity in the load path. Ramkumar and Tossavainen (77) evaluated the load eccentricity effects by comparing single lap test results with double lap test results. The results of the comparison indicated as much as 17 percent and 20 percent increase in the gross tensile and bearing strengths and the gross compressive and bearing strengths of the laminate, respectively, are obtained by changing from a single shear to a double shear configuration. Hart-Smith (76) compared the experimental results between single lap and double lap joints and found a 20 percent decrease with respect to double-shear strengths. He suggested that due account should be taken of the differences between single and double shear bolted joints in the analysis of practical aerospace structures.

3.3.6 Fastener Load Direction

Because of the anisotropic nature of composites, the bearing strength of composite laminates will vary with load direction. This effect was investigated by Garbo and Ogonowski (56) and by Matthews and Hirst (96). All material systems evaluated by these investigators were sensitive to load direction. Figure 23 illustrates the changes in circumferential stress distributions around a loaded fastener hole in a laminate with 70/20/10 lay-up as a result of shifting the direction of load from $\theta = 0°$ to $\theta = 45°$. The peak stresses no longer occur perpendicular to the load direction and the stress distribution is shifted and changed. The stress distribution for the isotropic case is included for comparison.

3.3.7 Failure Definition

For composite bolted joints, the determination of bearing strength depends on the definition of failure criterion which can vary widely from, simply, the maximum load sustained by the joint to a failure criterion based on the deformation of the hole. Johnson and Matthews (97), used typical load/extension plot of a composite bolted joint (see Figure 24), to suggest the following ways of defining failure load:

(a) The maximum load - Usually considerable damage will have occurred in reaching this load (81).

(b) The first peak in the load/extension plot - Damage sustained up to this load is not insignificant. Almost certainly cracks will have propagated outside of the washers (78).

(c) The load corresponding to a specified amount of hole elongation - There is little agreement as to what value should be used. Dastin (98), Strauss (99) and Oleesky and Mohr (100) use a value of 4 percent of the hole diameter, for glass reinforced plastics; Webb (101) uses 1 percent for bolts and 2 percent for rivets in CFRP; Althof and Muller (102) use 0.5 percent for CFRP; and Johnson and Matthews (97) use 0.4 percent of the original diameter for glass fiber-reinforced plastics. They suggested the limit load corresponding to 0.4 percent elongation of hole diameter could be obtained from the maximum load by using a factor of safety of 2.

(d) The load at which the load/extension curve first deviates from linearity -The point at which deviation from linearity occurs is usually difficult to establish. Also the slope of the load/extension curve may alter at more than one point (103).

(e) The load at which cracking first becomes audible - Johnson and Matthews (97) examined specimens at this point and found a few visible cracks around the loaded side of the hole.

(f) The load at which cracking is initiated - This load is probably quite low and very difficult to determine.

(g) The load at which cracks become visible outside the washers - Only one side of the specimen is accessible for visual detection.

Some of the loads defined above (d to g) are subject to wide variability or are difficult to determine objectively. The majority of investigators adopt the first three approaches (a to c) to define failure load.

Hole elongation and overall joint compliance criteria based on data obtained from load-deflection hysteresis curves are often used in fatigue tests to indicate the amount of cyclic damage accumulation and failure (56, 77).

3.4 Fatigue Test Results

Relatively few experimental results relating to fatigue behaviour of composite bolted joints are available in the published literature. A brief reference to fatigue of glass fiber-reinforced plastics joints is available in MIL-HDBK 17A (93). Reliability of composite joints is discussed by Wolff and Lemon (104) and by Wolff and Wilkins (105). In a special fastener development program for composite structures, Cole et al. (88) obtained test results using a typical tension-dominated fighter load spectrum for composite laminates mechanically joined by fasteners with four different flush-head configurations. The 100° tension head fastener was found to yield significantly improved fatigue performance over the 100° shear, 120° shear and 130° shear type fasteners.

Crews (109) investigated the bolt-bearing fatigue strength of graphite/epoxy (T300/5208) laminates. In his study, fatigue tests were conducted for a wide range of bolt clamp-up torques using single fastener coupons. The specimen with, w, and edge distance, e, were selected for w/d = 8 and e/d = 4. These ratios assured bearing failure rather than net-tension or shear-out failures. During all fatigue tests, the hole elongation due to cyclic loads was monitored until each specimen failed in the bearing mode. Ultrasonic C-scan methods were used to examine the area around the bolt hole for delamination damage due to cyclic loads. Experimental results indicated that bolt clamp-up had a beneficial effect on the fatigue limit which was improved by as much as 100 percent compared to the pin-bearing case. The explanation for this improvement was found to be the through-the-thickness compressive stresses provided by the clamp-up loads. These loads also influenced the amount of bolt hole elongations. It was found that high clamp-up torques virtually eliminated hole elongation during fatigue loading. C-scan images of specimens were obtained after 20,000 cycles of 87 ksi maximum bearing stress. At this stage, the presence of delamination damage around the hole was recorded. Also, a slight elongation of the hole was measured. This evidence indicated the initiation and growth of delaminations as a result of fatigue. The permanent hole elongation was caused by localized delaminations which formed a soft region around the hole. The original stiffness of the laminate in this region was reduced significantly. The further growth of the delamination under cyclic loads led to a bearing failure of the laminates. In his investigation, Crews did not attempt to establish any failure criterion for the bearing mode. It appears that existing models cannot predict the initiation, growth and instability as a result of delamination under cyclic loads in composite laminates.

An evaluation of critical joint design variables on fatigue life was carried out by Garbo and Ogonowski (56). Seven design parameters which have significant effects on the static strength of composite bolted joints were selected for evaluation. Tension-tension (R = +0.1), tension-compression (R = -1.0) constant amplitude testing and spectrum testing were performed using a pure bearing double-lap test specimen. Hercules AS/3501-6 graphite/epoxy was used to fabricate the test specimens. A review of the effects of the seven variables on fatigue performance based on the work of Garbo and Ogonowski is summarized below:

(1) Lay-up - Three laminate variations were tested to determine the relative fatigue life in terms of the number of fatigue cycles required to produce a 0.02 inch elongation which is about 5.3 percent of the hole diameter. For tension-tension cycling, the results indicated similar fatigue life for all lay-ups. For R = -1.0, the two matrix-dominant lay-ups (19/76/5 and 30/60/10) sustained fewer load cycles prior to developing a 0.02 inch hole elongation, as compared to the 50/40/10 fiber-dominant lay-up (Figure 25). For spectrum fatigue tests, the results showed no measurable hole elongation for any of the tested lay-ups after testing to an equivalent of 16,000 service hours.

(2) Stacking Sequence - The effects of stacking sequence were found to be insignificant with respect to joint fatigue life under both constant amplitude and spectrum cycling.

(3) Fastener Preload - Significant increases in fatigue strength were obtained by increasing the fastener preload from 0 inch pounds to 160 inch pounds. Failure modes were observed to be the same for all the fastener torque-up levels.

(4) Joint Geometry - The effects of specimen geometry on joint life were evaluated for the 50/40/10 and 19/76/5 lay-ups using specimens with different edge distances and widths. Fatigue strength was not changed for the 50/40/10 lay-ups. For the matrix-dominant 19/76/5 lay-up, fatigue strength was found to increase when the w/d ratio was increased from 3 to 4. Further, a change of failure mode from bearing to net section occurred when w/d ratio was reduced from 4 to 3.

(5) Interference Fit - Fatigue tests were conducted to evaluate the effect of an interference fit of 0.005 inch on the fatigue life of no-torque and torqued joints. For R = +.1, the results indicated that specimens with no preload and 0.005 interference fit failed at a lower fatigue life relative to neat-fit specimens with no preload. Associated with this lower life was a large data scatter. Standard pull-through installation techniques and fastener types were used. Some joint specimens were sectioned and examined. The presence of delamination at the fastener exit side as well as through-the-thickness of the hole in the laminate were discovered for joints with interference fits from .004 through .007 inch. Therefore, it was suspected that minor installation damage caused by inserting the interference fit fastener produced the wide scatter in data when no clamping constraint existed due to zero fastener preload. Limited fatigue testing of specimens with 160 inch pounds preload and 0.005 inch interference fit produced higher fatigue lives relative to neat-fit specimens with the same installation torque.

(6) Single-Shear Loading - The effect of bending stresses due to the single shear configuration resulted in a significant reduction in fatigue life. Further fatigue life reductions were exhibited by the countersunk fastener with its associated loss of direct bearing material and added fastener head flexibility relative to the protruding head fastener. Joint spring rates for the double-shear, non-countersunk single-shear, and countersunk single-shear specimens were found to be 396,000 lbs/in., 347,000 lbs/in., and 250,000 lbs/in. respectively.

(7) Porosity - Tests of specimens with moderate porosity resulted in no reduction in either static strength or joint fatigue life.

4.0 SUMMARY

Technical observations pertaining to composite mechanically fastened joint technology as a result of this review are summarized in the following:

(1) Similar directions have been followed throughout the technical community in the analysis of composite bolted joints in aircraft structural components. The analysis proceeds from an overall structural analysis, to localized joint idealization and bolt-load distribution analysis, and finally to an assessment of the static and/or fatigue strength through the utilization of joint failure analysis at individual fastener holes.

(2) Static joint failure analysis for composites, which represents the primary area of research activity in the literature reviewed, consists of detailed stress analysis performed at individual fastener holes and associated application of a failure criterion.

(3) State-of-the-art stress analysis methodologies include finite element, two-dimensional anisotropic elastic analysis and fracture mechanics. Three-dimensional analysis is needed for determining the through-the-thickness stress distribution around fastener holes in order to assess the role of bearing and interlaminar shear failures.

(4) Static strength is evaluated by applying anisotropic material failure criteria after the stresses at a "characteristic dimension" from the edge of the hole are determined. Quadratic failure surfaces, which account for failure mode interaction, have been demonstrated to be capable of predicting both failure load and failure mode accurately in composite bolted joints for most cases. A cubic formulation has been shown to yield more accurate predictions in biaxial loading cases.

(5) State-of-the-art fatigue life prediction methodologies require an extensive data base which, except for some specific cases, is not well established for composites. Predictions based on Miner's linear cumulative damage model have been found to be unconservative. The assumption of continuously decreasing residual strength in the residual strength degradation model restricts the model from general applications. "Fatigue function" models have achieved some preliminary success in correlating with experimental results. These models have been extended to include delamination and temperature effects. Empirical approaches are most common in predicting fatigue life of composites as well as in demonstrating compliance with military and commercial durability specifications.

(6) The effects of various design parameters on the static and fatigue behaviour of composite bolted joints have been examined. For quasi-isotropic lay-ups, minimum allowed edge distances and fastener spacings ranged respectively from 3-4 and 4-5 times fastener diameters in most design practices where full bearing strength is required of the joint. In general, the tensile and bearing strengths decrease with increasing hole diameter and decreasing laminate thickness. Load eccentricity in a single lap joint reduces both

the static and fatigue strength significantly. Due to the anisotropic nature of composites, the bearing strength has been shown to vary with fastener load direction.

(7) Galvanic corrosion, galling, installation damage and low pull-through strength are recognized as the four basic problems where mechanical fastening of composite laminates is concerned. In general engineering practices, only tension head fasteners made of titanium are used with composites and no interference fit holes, hole filling fasteners, or vibration driving of rivets are recommended. Special fastener systems have been developed to allow for interference installation using appropriate toolings to minimize installation damage. It has been demonstrated that interference fit increases fatigue strength. The application of bolt torque to the optimum level has been accomplished by using fasteners with large "foot-print" or washers. Application of special lubricants and proper material selection eliminate galling problems. In general, high clamping forces increase static and fatigue strength of a composite bolted joint.

(8) There is currently no bolted joint data readily available for the advanced composite materials.

5.0 RECOMMENDATION - FURTHER RESEARCH

As a result of this review, the need for further research in the following areas is identified:

(1) The accuracy of the analytical prediction of the static strength of mechanically fastened joints depends strongly on the stress analysis and the failure criterion applied. In order to improve the accuracy of the strength prediction, further work is required in the three dimensional stress analysis of a joint to account for the through-the-thickness effects as a result of interference fit and preload. It is also deemed necessary to incorporate an elastic contact analysis with stick-slide behaviour in the three dimensional analysis in order to provide more sophisticated solutions which include frictional effects. Further work in the development of failure criteria based on physical damage phenomena to predict delamination and gross bearing failure modes is required.

(2) Further research is required in the development of fatigue life prediction methodologies. It appears that static strength prediction is sufficient in most cases because of the relative insensitivity of composite materials to fatigue. However, the initiation and growth of delaminations have been observed to take place under cyclic loading at free edges, especially at the edge of a fastener hole because of the stress concentration. This delamination creates a local region near the bolt hole where the original stiffness of the composite laminate has been reduced extensively. Consequently,

the growth of delaminations can cause extensive elongation of the fastener hole. Further damage can be accumulated rapidly as a result of the pounding of a loose fastener in a hole when the load spectrum includes reversals ($R < 0$). This can lead to the ultimate loss of the overall load-carrying capability of the composite component as a result of an extensive drop in stiffness or static strength. Further research is deemed necessary since methodologies available currently cannot predict initiation, growth and instability as a result of delamination under fatigue loading in composites.

(3) A critical evaluation is deemed necessary to compare and assess the reliability aspects of various existing fatigue life prediction methodologies and establish the specific conditions under which they are reliable. These specific conditions include: (a) operating conditions, e.g., temperature, environment, loading; (b) failure modes, e.g., delamination, bearing; (c) material parameters, e.g., ply orientation; and (d) specific assumptions, e.g., plane stress, continuously decreasing residual strength. This evaluation may provide a basis for the selection of methodologies to predict the fatigue life of composite mechanically fastened joints reliably under the conditions of interest.

(4) The application of advanced composite material systems should be investigated. These systems include high strain carbon fibers and tough resin matrix such as bismaleimide. Specifically, bearing properties and stress concentration sensitivity should be studied. Furthermore, the effects of the improved interlaminar normal and shear properties as a result of the tougher resin matrix on the clamping load and interference fit provided by the fastener systems should be investigated because, as indicated by the present literature review, enhanced static and durability performances of composite mechanically fastened joints could be obtained by increasing the clamping load and interference fit.

(5) No published data was found on the use of mechanical fasteners in high strain/tough resin composites. In order to design mechanically fastened joints using these high strain/tough resin composites, experimental work should be undertaken to generate design data similar to those generated based on conventional graphite/epoxy composites. The design data generated can be used to verify/establish the benefits of using these systems. For example, weight savings are predicted as a result of the improvement in the allowable design strain level provided by these advanced composites. Also, data is required to examine the applicability of the current analysis procedures and failure criteria based on conventional graphite/epoxy systems to predict the static and fatigue strengths and failure modes of mechanically fastened joints fabricated from these advanced composites.

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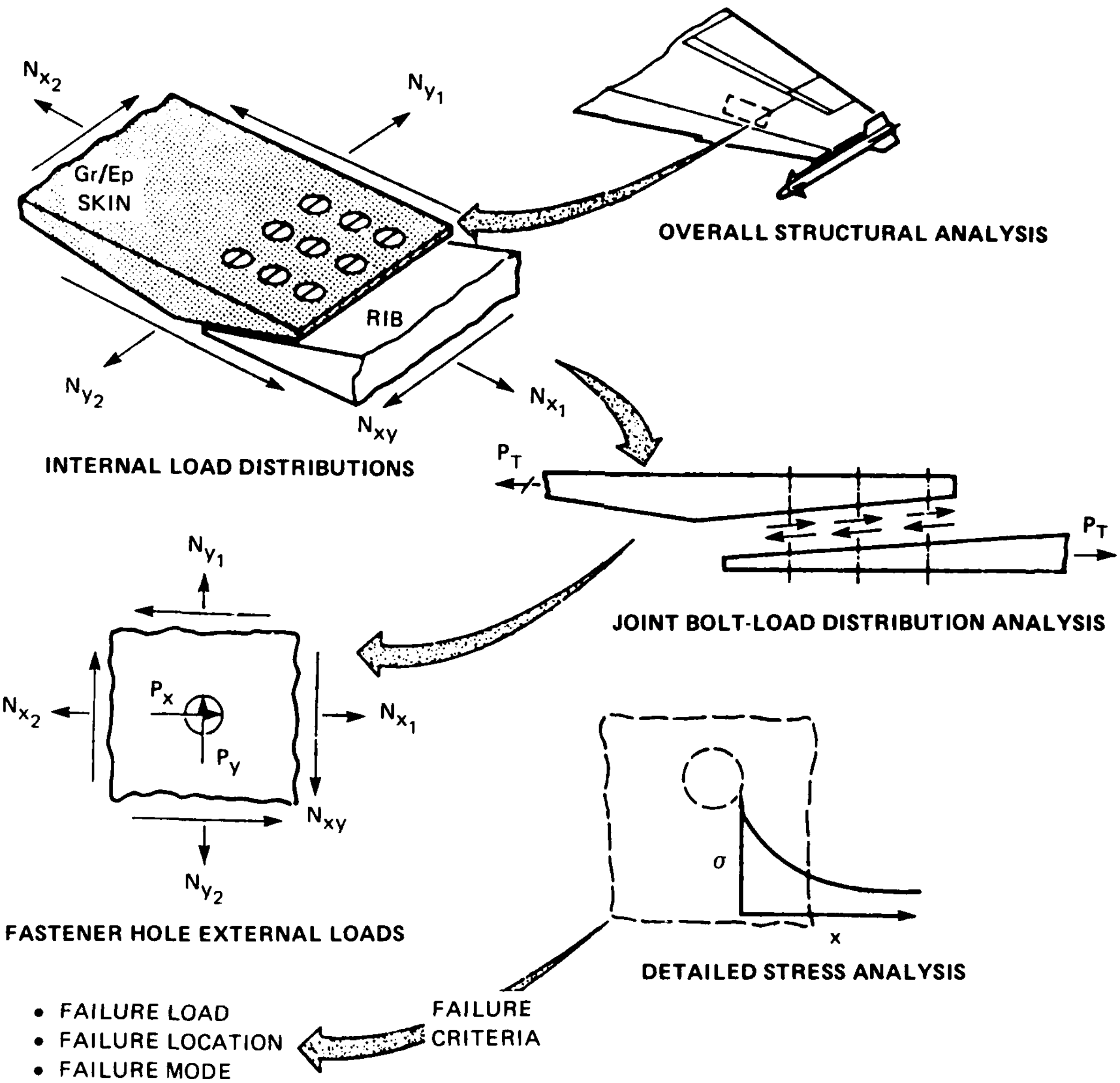


FIG. 1: JOINT FAILURE ANALYSIS (REF. 1)

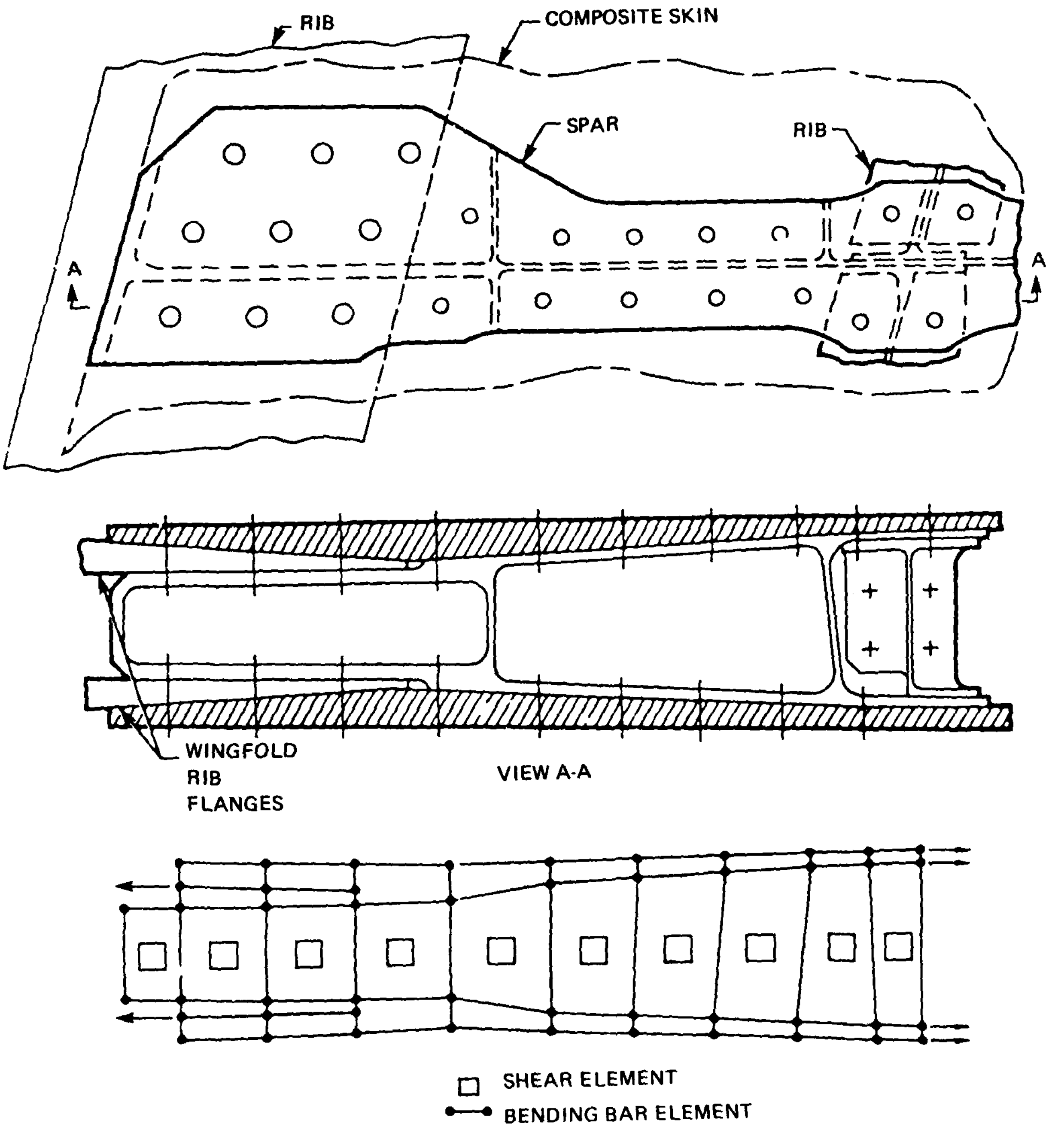
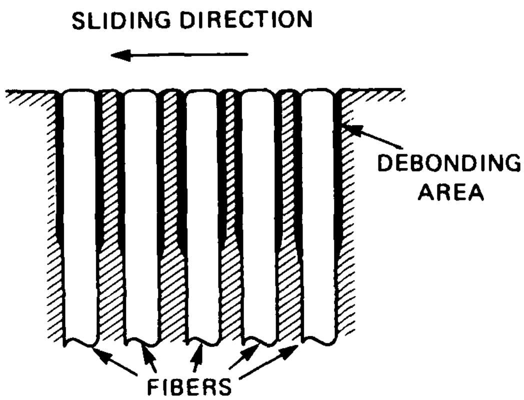
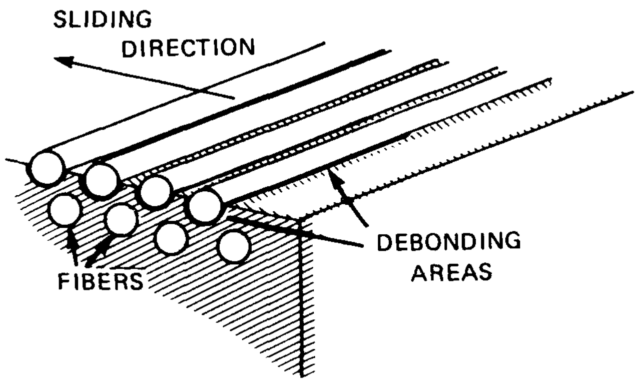


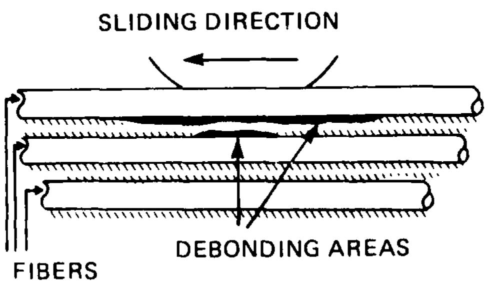
FIG. 2: WINGFOLD SPLICE JOINT IDEALIZATION (REF. 1)



(a) **FIBER ORIENTATION NORMAL TO SLIDING DIRECTION**



(b) **FIBER ORIENTATION TRANSVERSE TO SLIDING DIRECTION**



(c) **FIBER ORIENTATION PARALLEL TO SLIDING DIRECTION**

FIG. 3: SCHEMATIC REPRESENTATION OF FAILURE MODES IN UNIAXIAL CONTINUOUS FIBER REINFORCED COMPOSITE UNDER SLIDING SURFACE FIBER ORIENTATION NORMAL (a), TRANSVERSE (b) AND LONGITUDINAL (c) TO THE SLIDING DIRECTION (REF. 11)

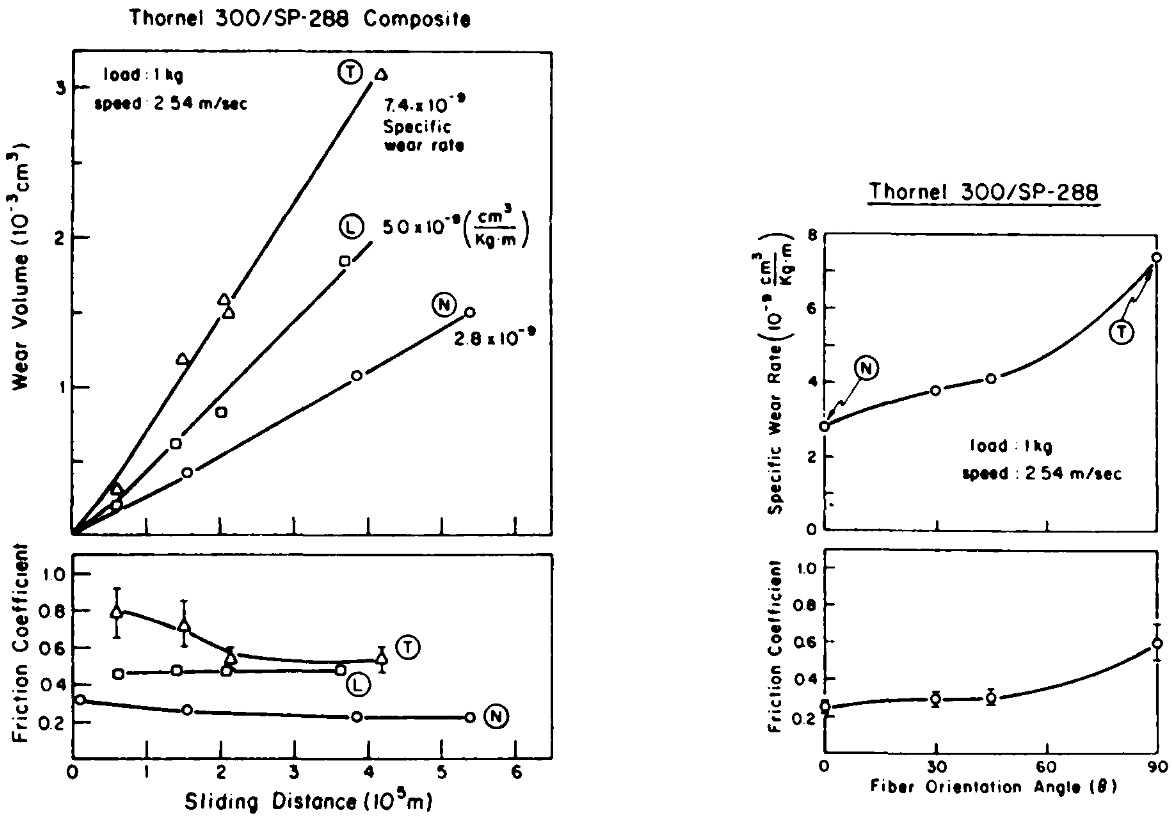


FIG. 4: FRICTION COEFFICIENTS AND WEAR VOLUME AS A FUNCTION OF SLIDING DISTANCE IN UNIAXIAL GRAPHITE FIBER–EPOXY COMPOSITE. SLIDING AGAINST 52100 STEEL, WITH FIBER ORIENTATIONS NORMAL, LONGITUDINAL AND TRANSVERSE TO THE SLIDING DIRECTION. (REF. 11)

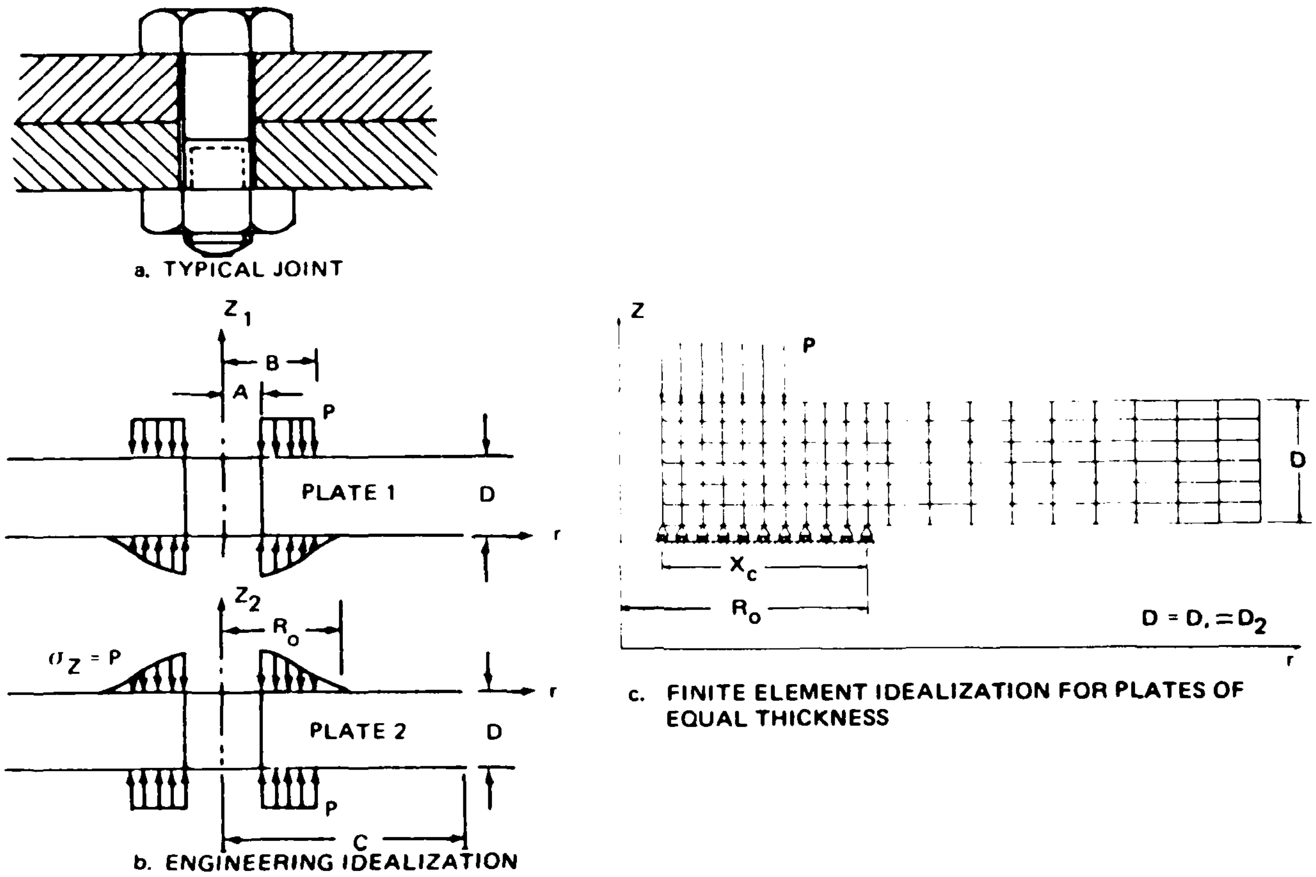


FIG. 5: IDEALIZATION OF TIGHTLY BOLTED JOINT TO DETERMINE CONTACT PRESSURE BETWEEN PLATES OF EQUAL THICKNESS (REF. 12)

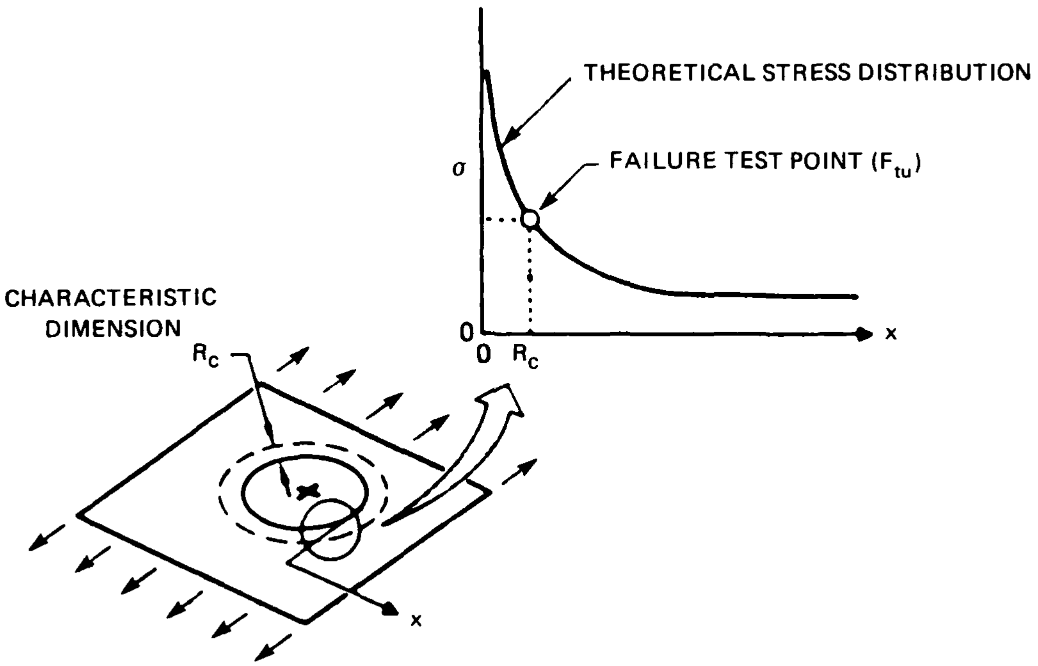


FIG. 6: FAILURE HYPOTHESIS (REF. 18)

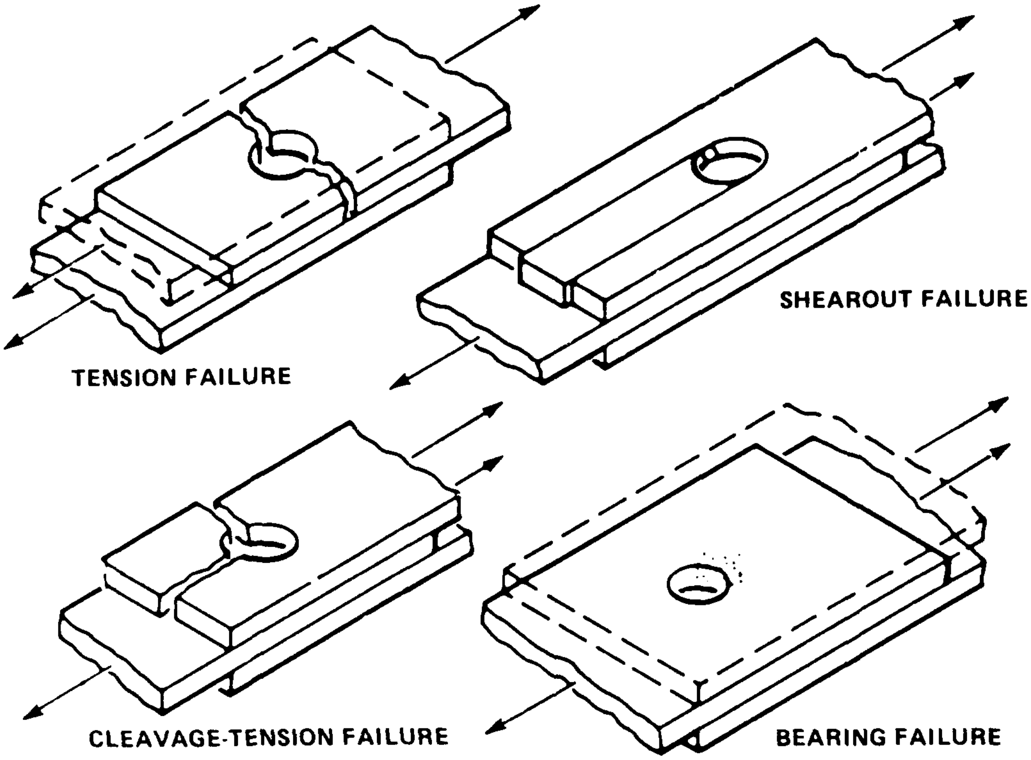
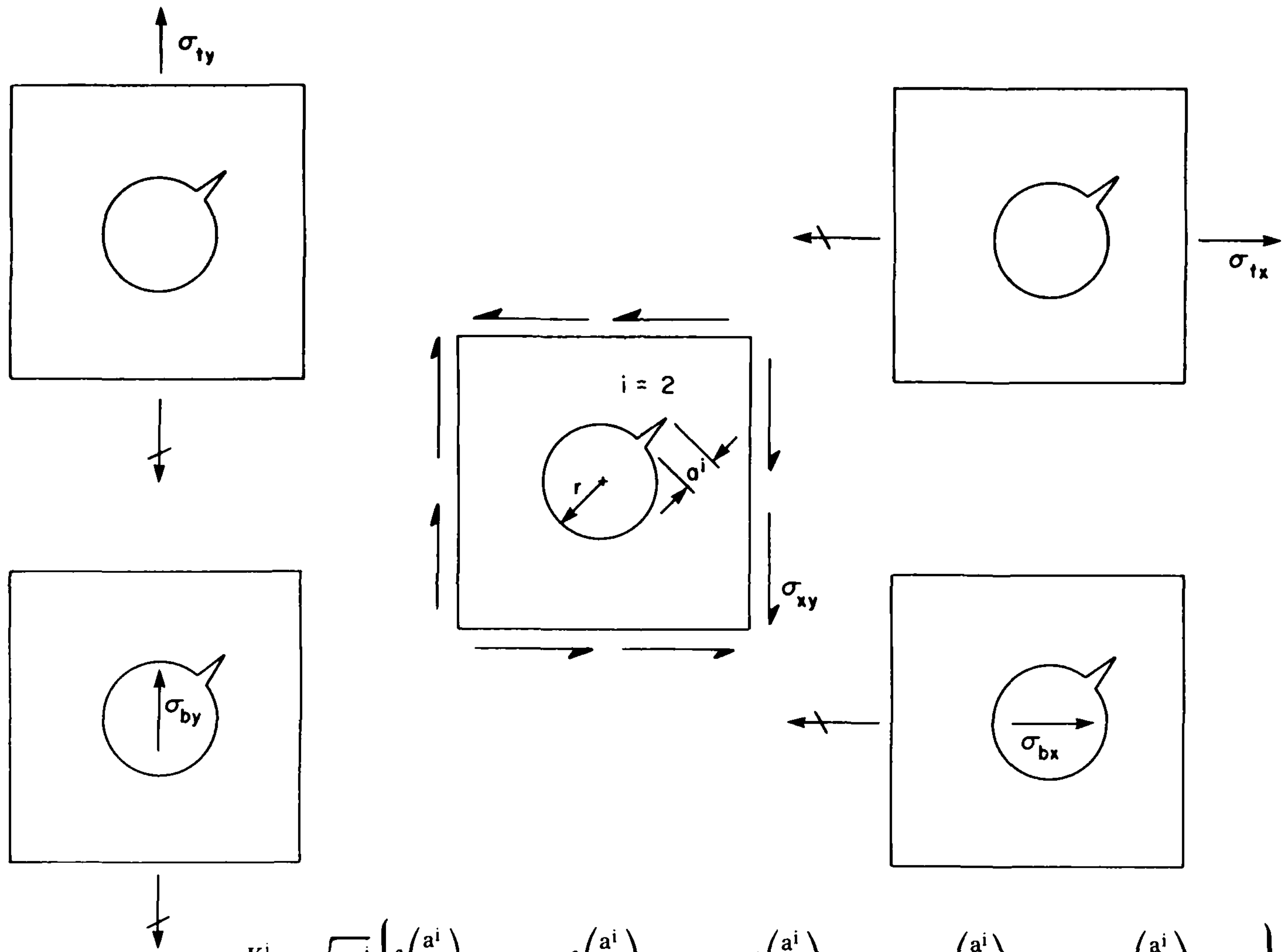


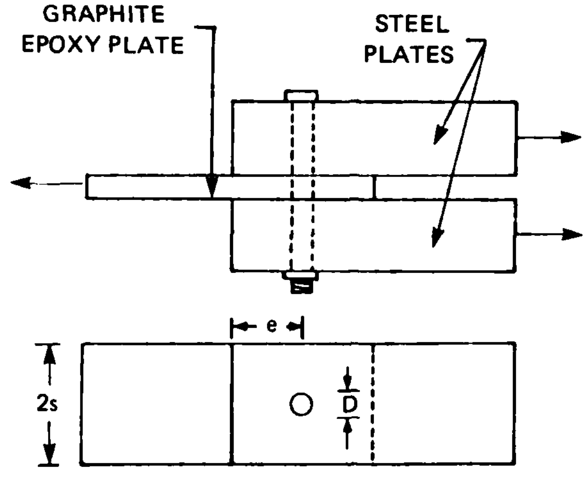
FIG. 7: MODES OF FAILURE FOR BOLTED COMPOSITE JOINTS



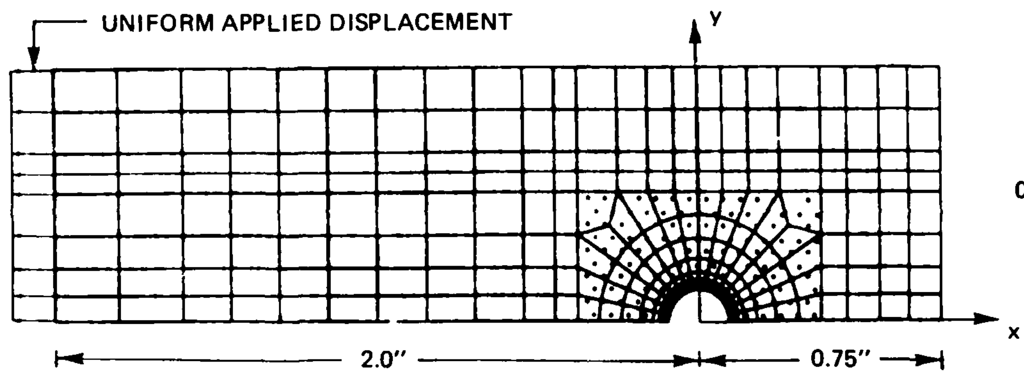
$$K_I^i = \sqrt{\pi a^i}\left\{ f\left(\frac{a^i}{r}\right)_{tx} \sigma_{tx} + f\left(\frac{a^i}{r}\right)_{ty} \sigma_{ty} + f\left(\frac{a^i}{r}\right)_{xy} \sigma_{xy} + f\left(\frac{a^i}{r}\right)_{bx} \sigma_{bx} + f\left(\frac{a^i}{r}\right)_{by} \sigma_{by} \right\}$$

FAILURE CRITERION: $K_I^i = K_Q^i$

FIG. 8: COMPUTATION OF STRESS INTENSITY FACTOR USING LINEAR SUPERPOSITION (REF. 32)



SKETCH OF THE BOLTED JOINT SETUP



NASTRAN MODEL FOR BOLT BEARING SPECIMEN

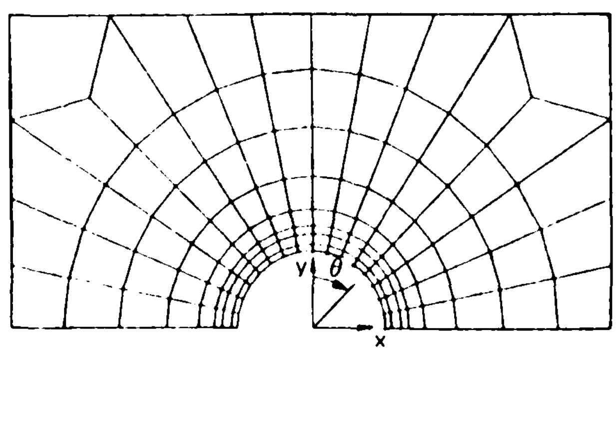
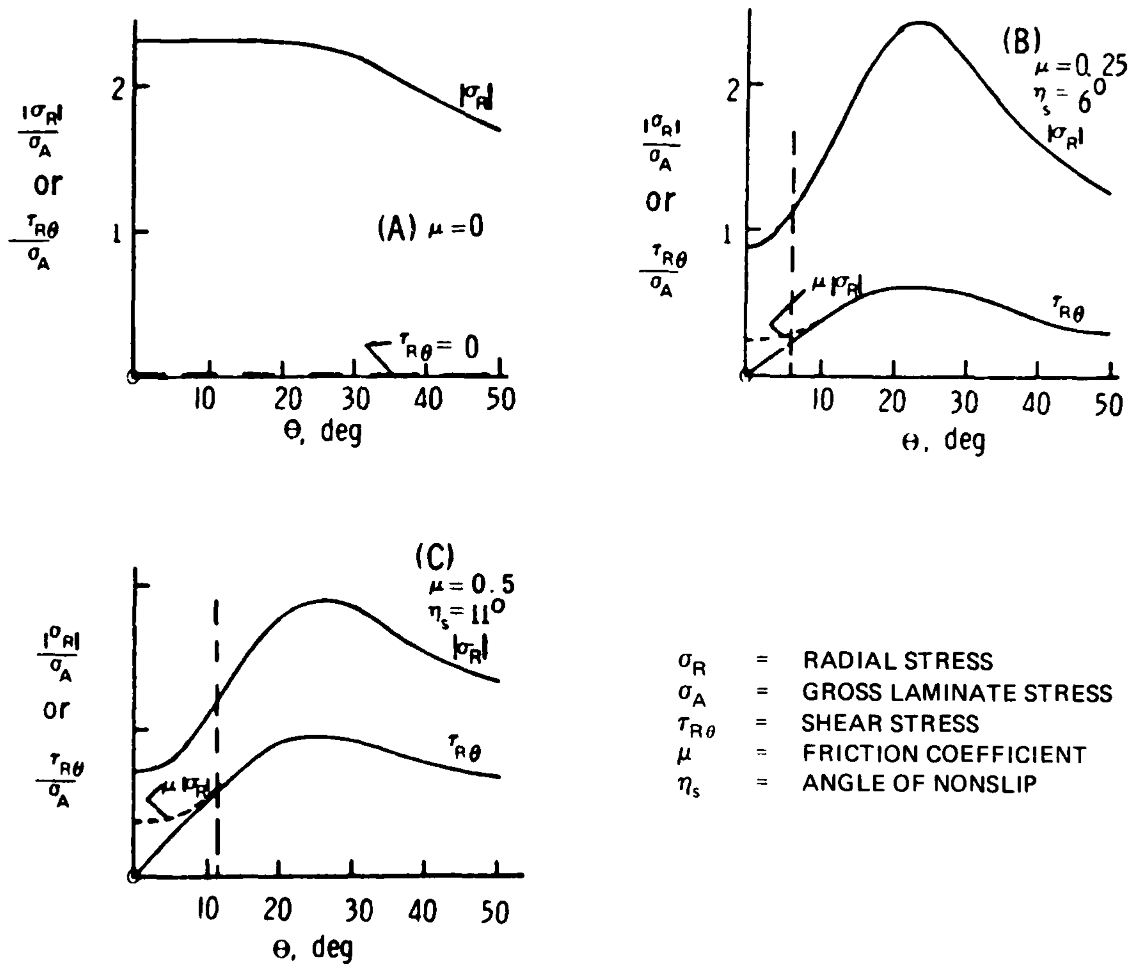


FIG. 9: NASTRAN MODEL OF THE REGION IN THE VICINITY OF THE FASTENER HOLE (REF. 39)



σ_R = RADIAL STRESS
σ_A = GROSS LAMINATE STRESS
$\tau_{R\theta}$ = SHEAR STRESS
μ = FRICTION COEFFICIENT
η_s = ANGLE OF NONSLIP

FIG. 10: EFFECT OF FRICTION ON RADIAL AND SHEAR STRESS DISTRIBUTION AROUND FASTENER HOLE, $0_2 \pm 45$ GRAPHITE EPOXY, e/D = 4, s/D = 1 (MULTI-PIN) (REF. 59)

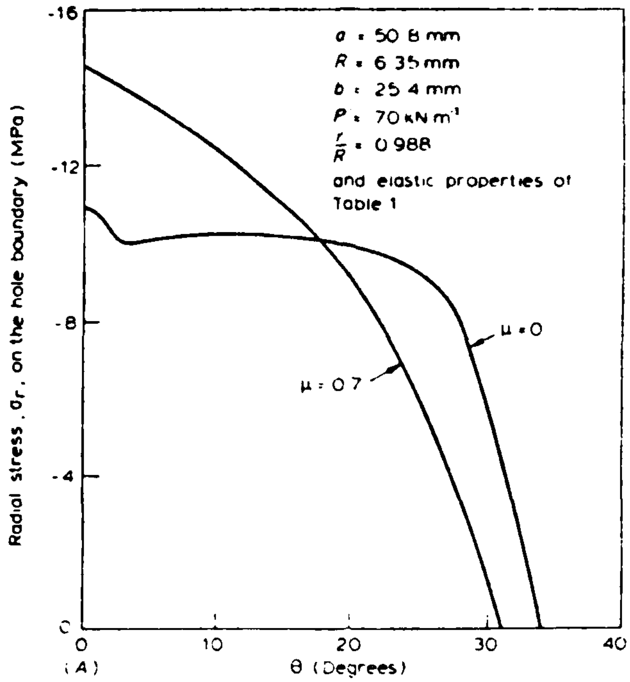


FIG. 11: EFFECT OF FRICTION ON THE RADIAL STRESS BETWEEN BOLT A AND WOOD OF A SINGLE FASTENER IN SITKA SPRUCE (REF. 48)

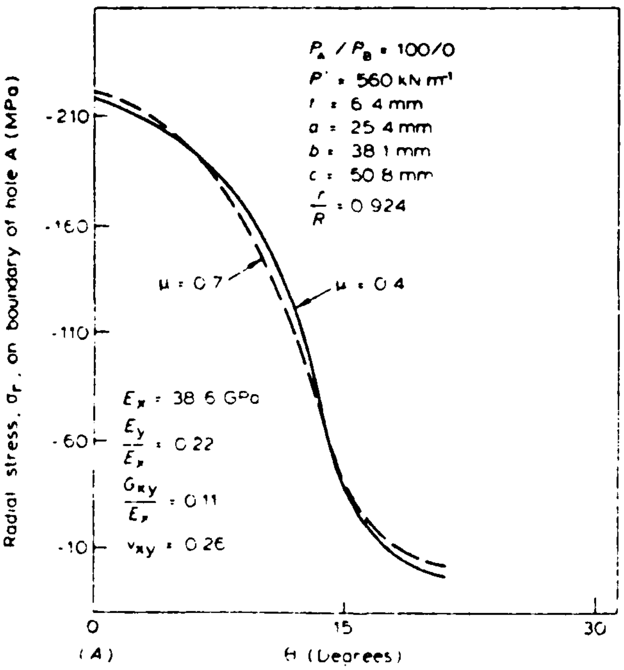


FIG. 12: THE RELATIVE INSENSITIVITY TO VARIATIONS IN FRICTION OF THE RADIAL STRESS BETWEEN BOLT A OF A DOUBLE FASTENER JOINT AND THE CONTACTING FIBREGLASS COMPOSITE (REF. 48)

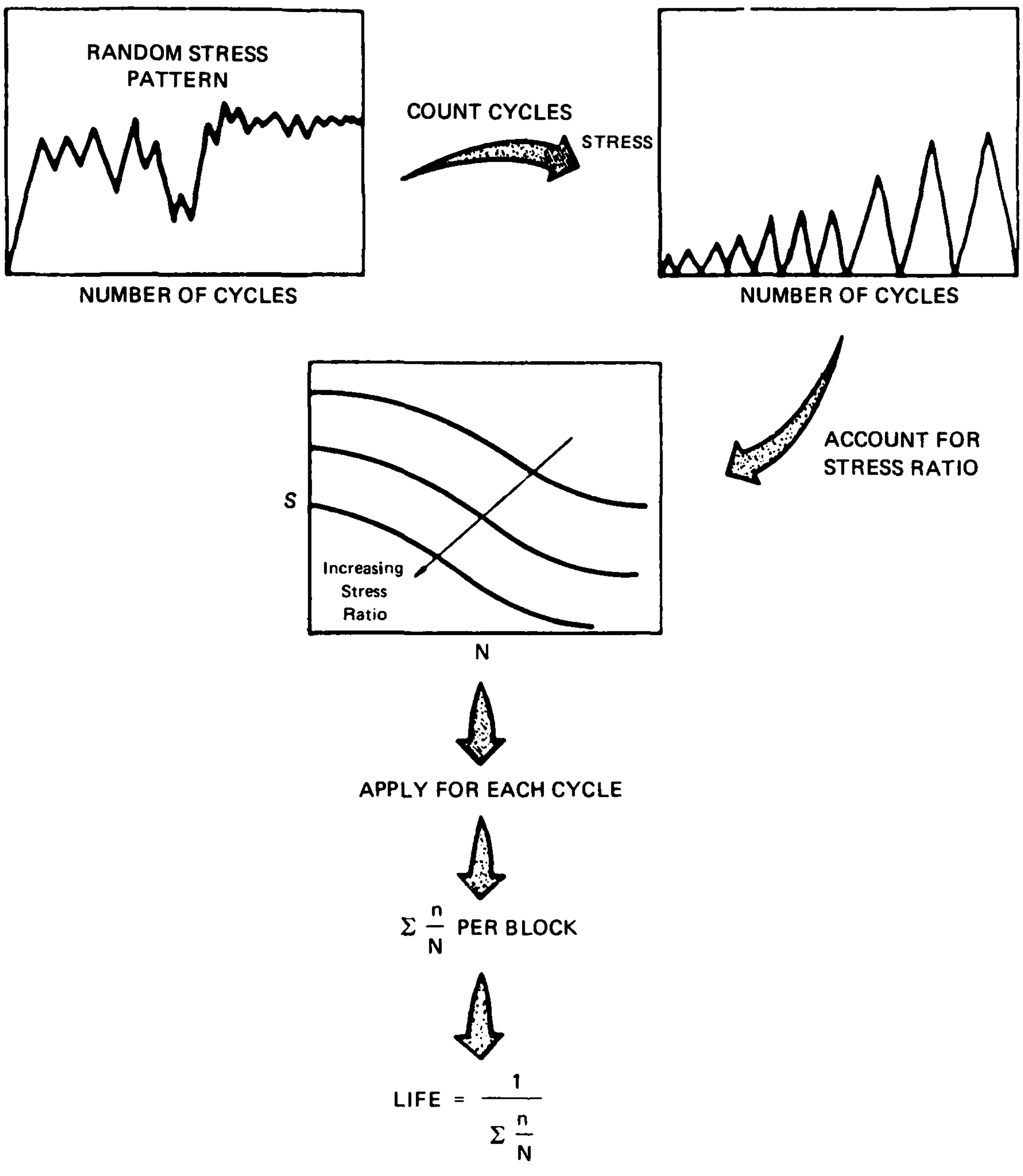


FIG. 13: MINER'S RULE APPLIED TO COMPOSITES

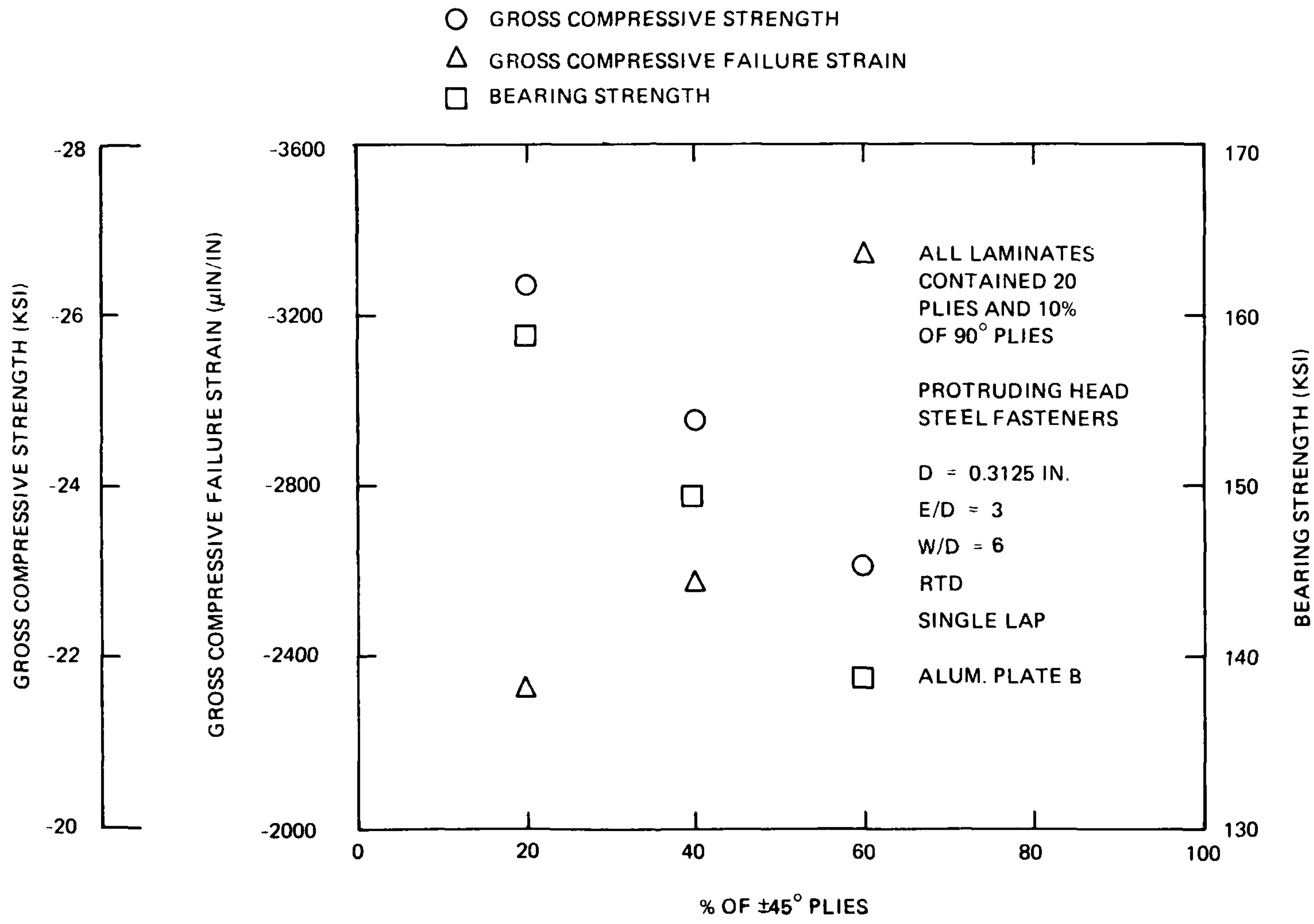


FIG. 14: EFFECT OF LAYUP ON THE COMPRESSIVE RESPONSE OF 20-PLY LAMINATES (REF. 77)

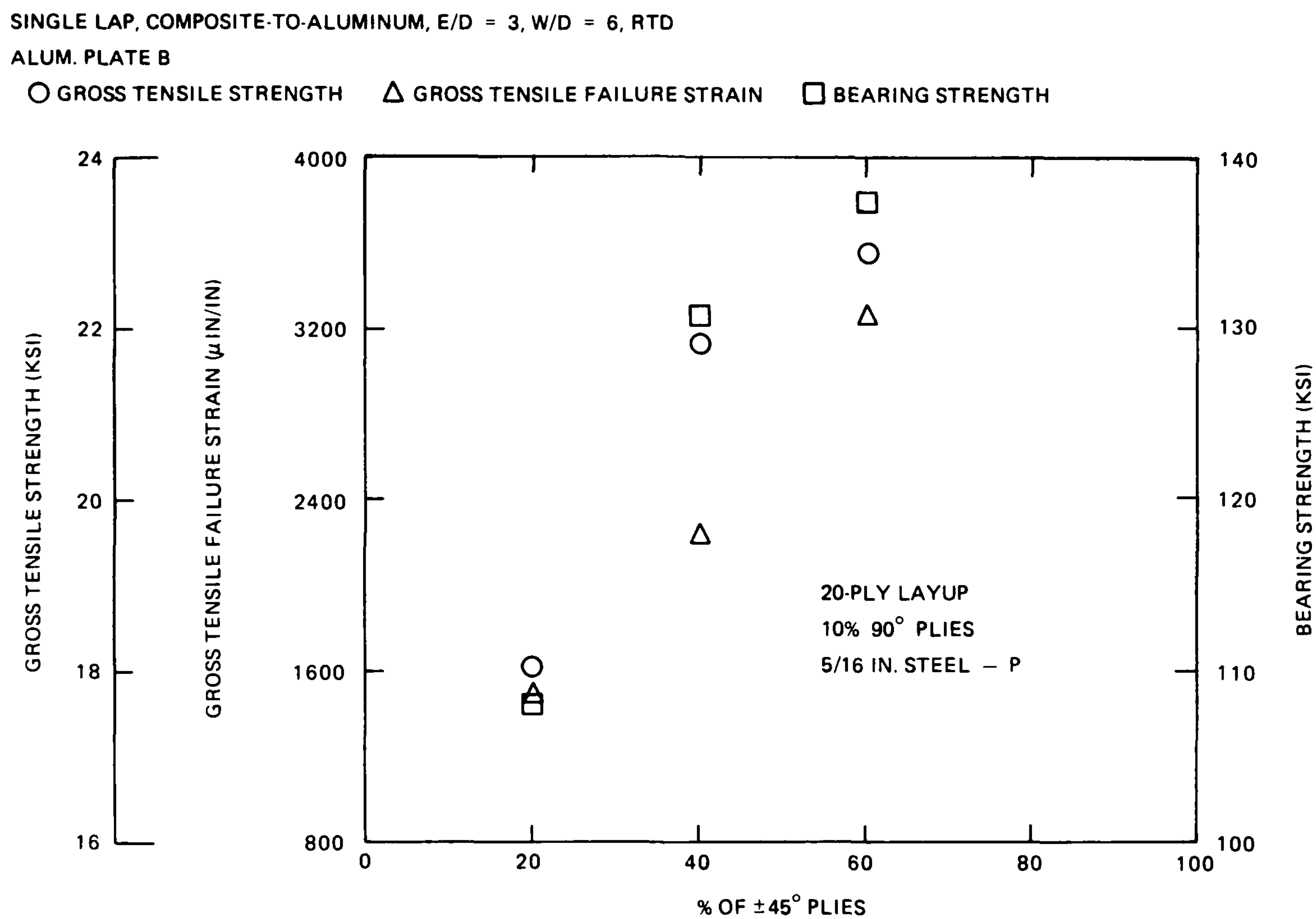
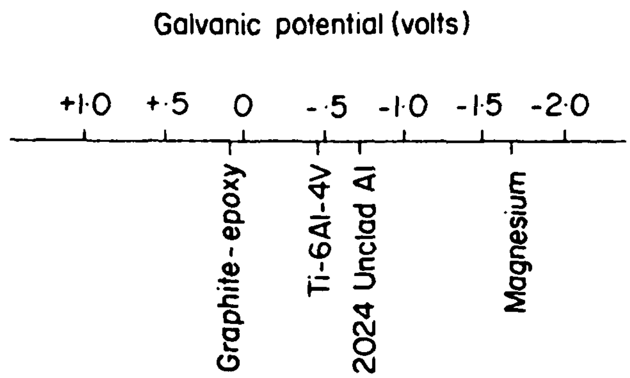


FIG. 15: EFFECT OF LAMINATE LAYUP ON THE TENSILE RESPONSE OF 20-PLY LAMINATES IN SINGLE SHEAR (REF. 77)

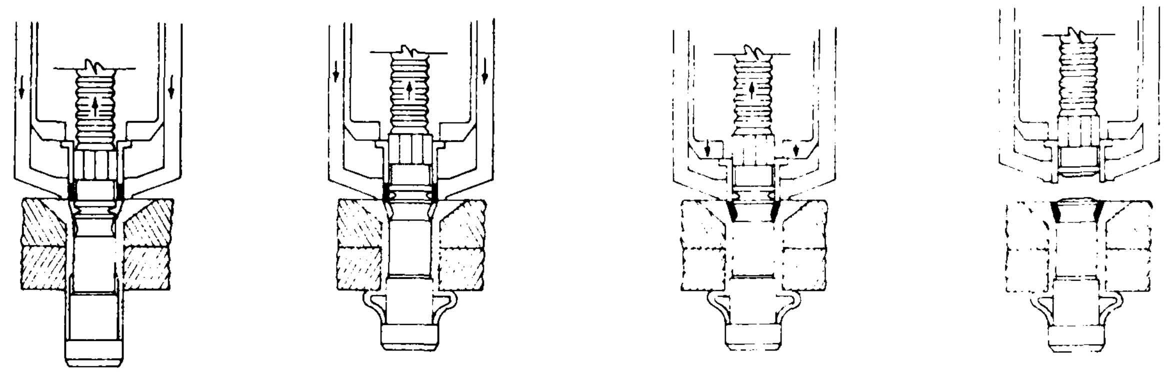


Comparison of galvanic potential of structural materials

Preference No	Material	Compatibility with graphite/epoxy
1	Titanium, Ti alloys, Ti-Nb	Compatible
2	MP-35N, Inco 600	Compatible
3	A286, PH13-8Mo	Marginally acceptable
4	Monel	Marginally acceptalbe
5	Low alloy steel; martensitic stainless steel	Not Compatible

Summary of NASC galvanic compatibility chart

FIG. 16: GALVANIC CORROSION RESULTS FROM DIFFERENCES IN POTENTIAL (REF. 88)



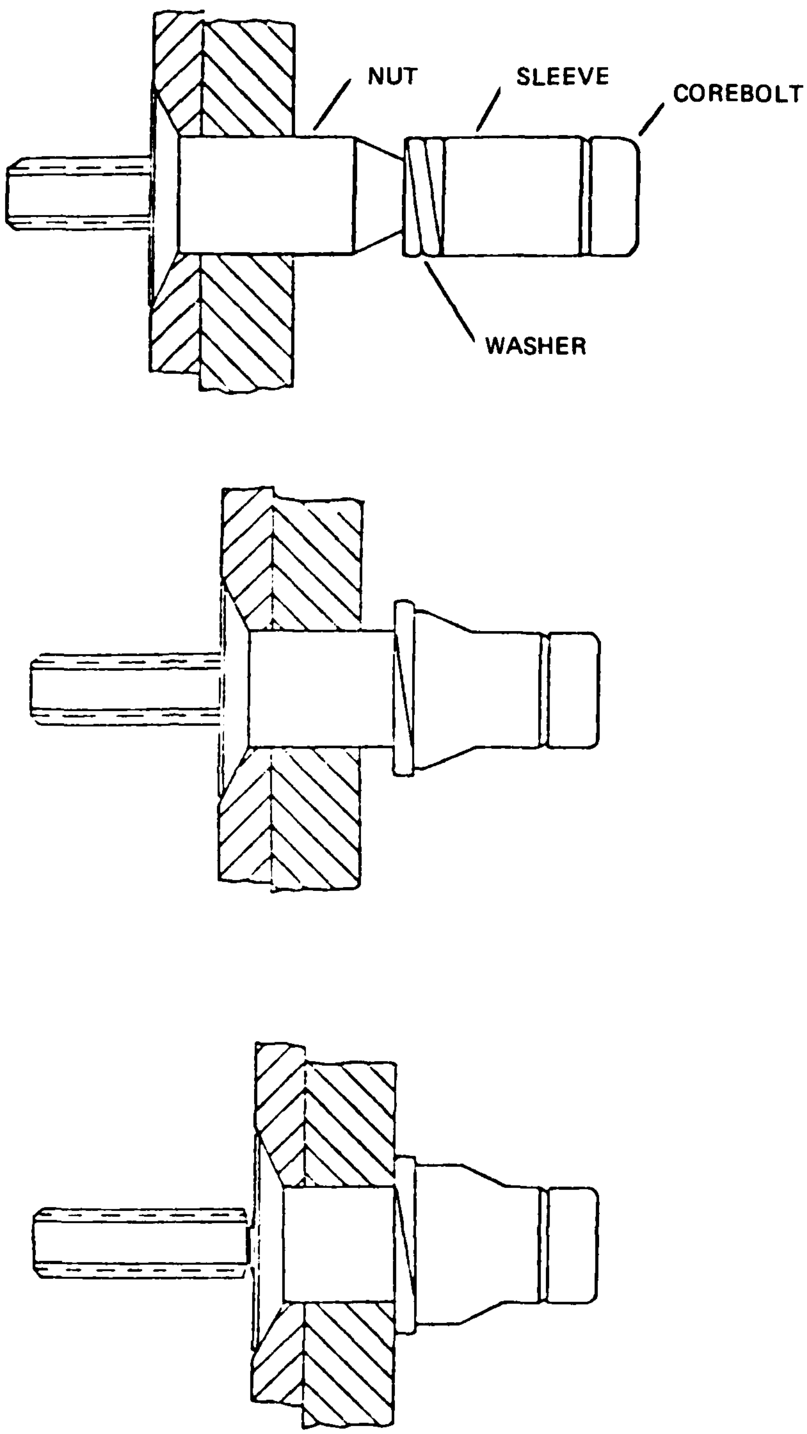
Step 1. During the initial part of the driving operation, the sleeve is squeezed between the head of the pin and the nose of the rivet tool.

Step 2. The head of the pin upsets the sleeve to form a strong, bulbed head on the blind side.

Step 3. When the blind head has been formed, the tool automatically forces the locking collar (at the pintail end of the sleeve) into the conical space between the recess in the sleeve head and the locking groove in the pin. This locks the parts together permanently.

Step 4. Pin is broken off in tension at the breakneck groove, substantially flush with the head of the sleeve. There is no projecting pin left to be cut off in a separate operation.

FIG. 17: INSTALLATION SEQUENCE OF A TYPICAL BLIND FASTENER FOR SOLID COMPOSITE LAMINATES



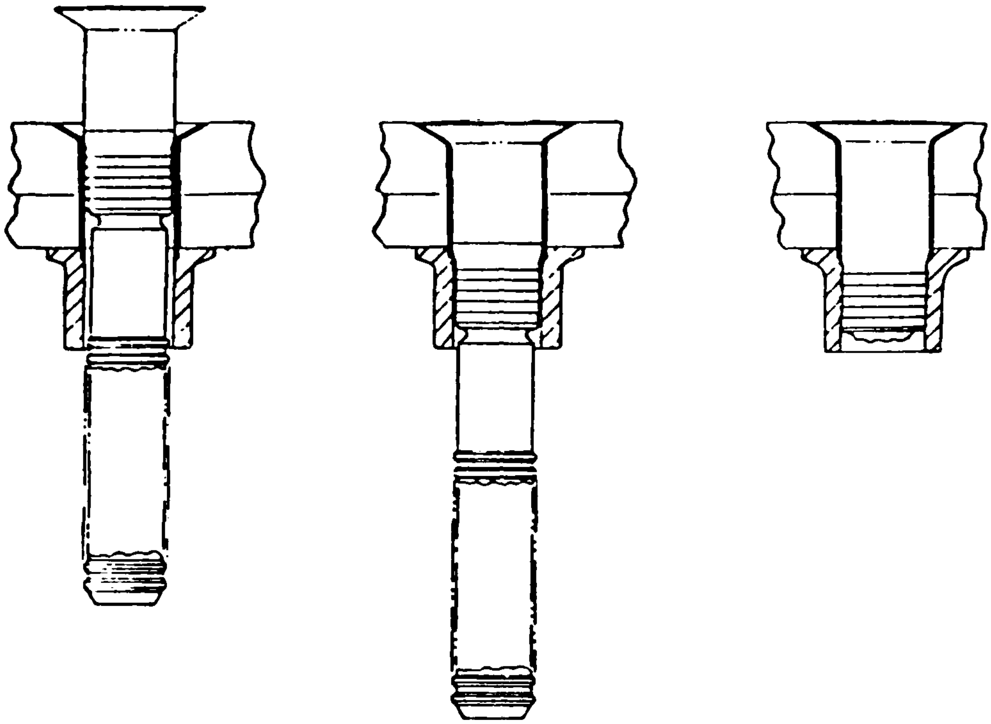
Using NAS 1675-type installation tooling, the nut is restrained from turning while the corebolt is driven.

The advance of the corebolt forces the washer and sleeve over the taper, expanding and uncoiling the washer to its maximum diameter.

Continued advance of the corebolt draws the washer and sleeve against the joint surface, preloading the structure.

At a torque level controlled by the break groove, the slabbed portion of the corebolt separates, and installation is complete.

FIG. 18: COMP-TITE™ BLIND FASTENER INSTALLATION SEQUENCE



CIL INSTALLATION PROCEDURE

STEP 1. PLACE ASSEMBLY IN WORK STRUCTURE. INSERT COLLAR OVER PINTAIL. (GRIPS THRU 2D ONLY. SEE NOTE BELOW)

STEP 2. TOOL PULLS ON PINTAIL AND DRAWS THE CIL PIN INTO THE SLEEVE WHICH EXPANDS THE SLEEVE TO COMPLETELY SEAL THE FASTENER IN THE WORK.

STEP 3. AS THE PULL ON THE PIN INCREASES, TOOL ANVIL SWAGES COLLAR INTO LOCKING GROOVES AND A PERMANENT LOCK IS FORMED. TOOL CONTINUES TO PULL UNTIL THE PIN BREAKS AT THE BREAK-NECK GROOVE AND IS EJECTED. TOOL ANVIL DISENGAGES FROM SWAGED COLLAR.

NOTE: ABOVE 2D, IT IS NECESSARY TO PULL THE PIN/SLEEVE ASSEMBLY INTO THE WORK WITH A SPECIAL PULL IN TOOL. WHEN SEATED PUT THE COLLAR OVER THE PINTAIL AND INSTALL THE FASTENER USING THE STANDARD INSTALLATION TOOL. SEE ABOVE CHART FOR NOSE INSTALLATION INFORMATION AND PROCEDURES.

FIG. 19: INSTALLATION SEQUENCE OF A TWO-PIECE LOCKBOLT FASTENER FOR SOLID COMPOSITE LAMINATES

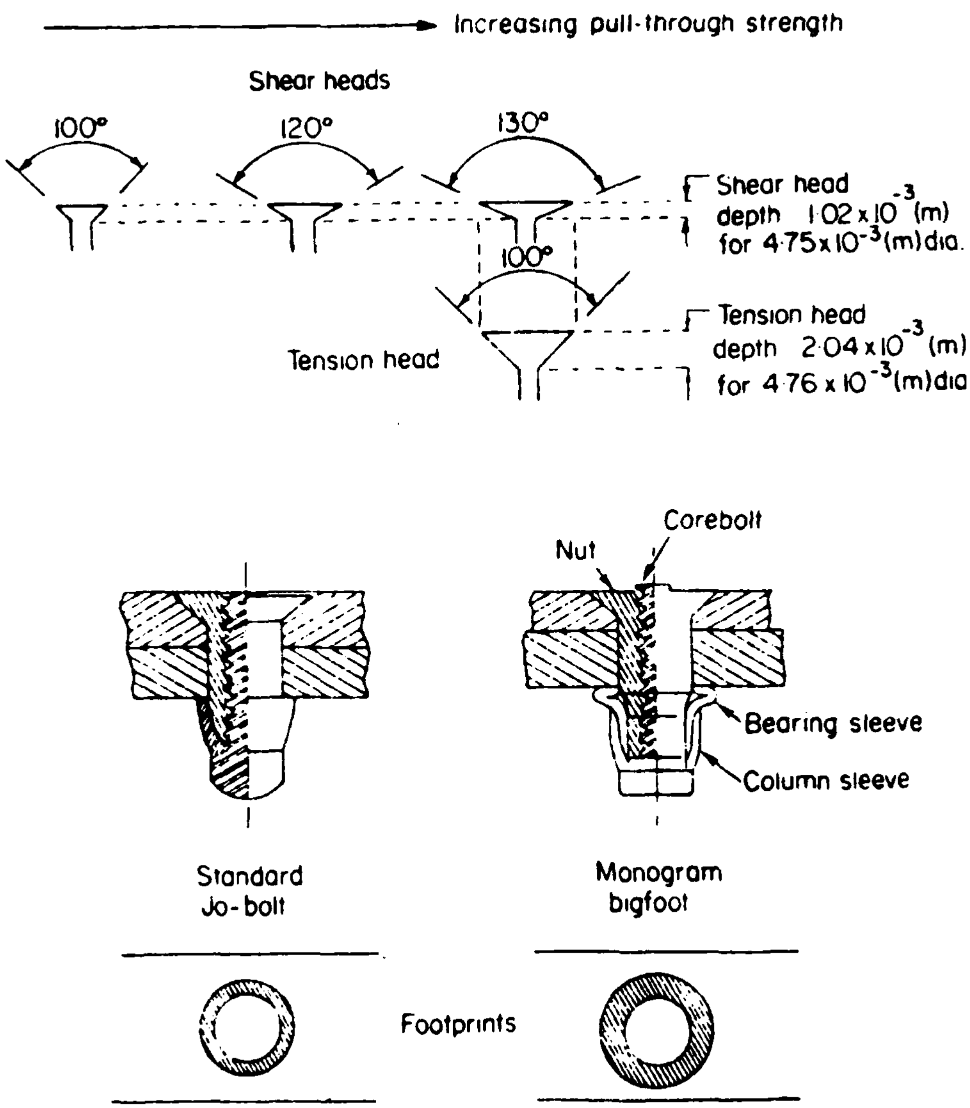


FIG. 20: LARGER FLUSH HEAD DIAMETER AND FOOTPRINT INCREASE PULL-THROUGH STRENGTH (REF. 88)

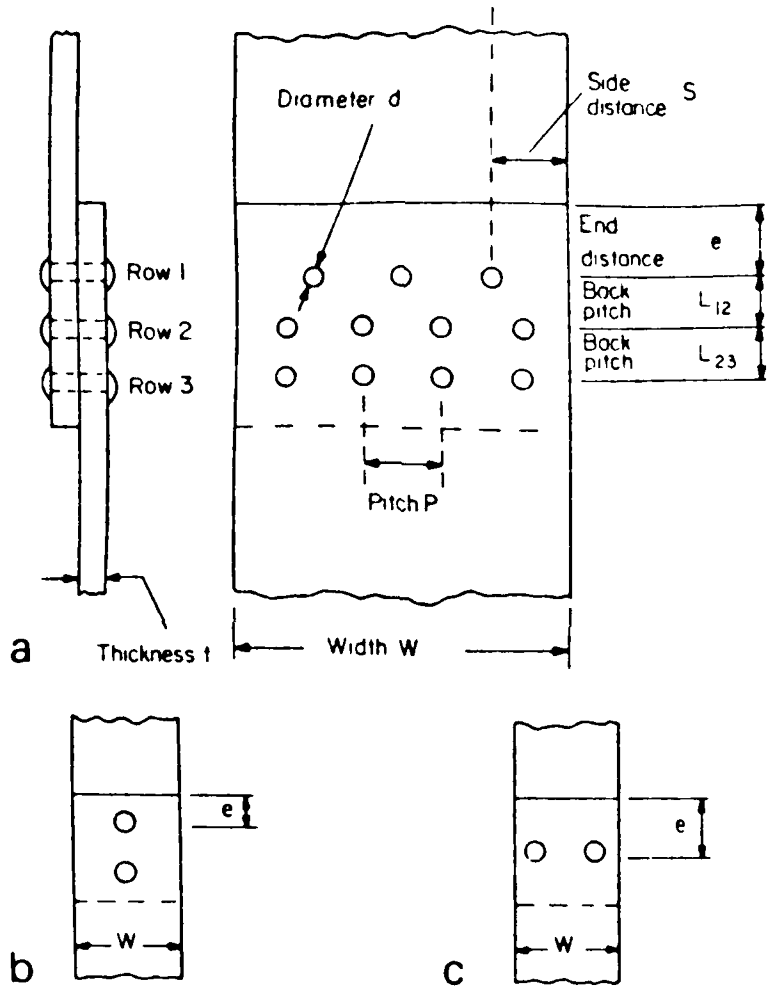


FIG. 21: DEFINITION OF JOINT GEOMETRY: A) ROWS OF RIVETS (ROWS 1 AND 2 ARE STAGGERED, ROWS 2 AND 3 ARE IN UNIFORM RECTANGULAR PATTERN); B) 2 RIVETS IN TANDEM (LINE OF RIVETS); C) 2 RIVETS IN PARALLEL (ROW OF RIVETS) (REF. 92)

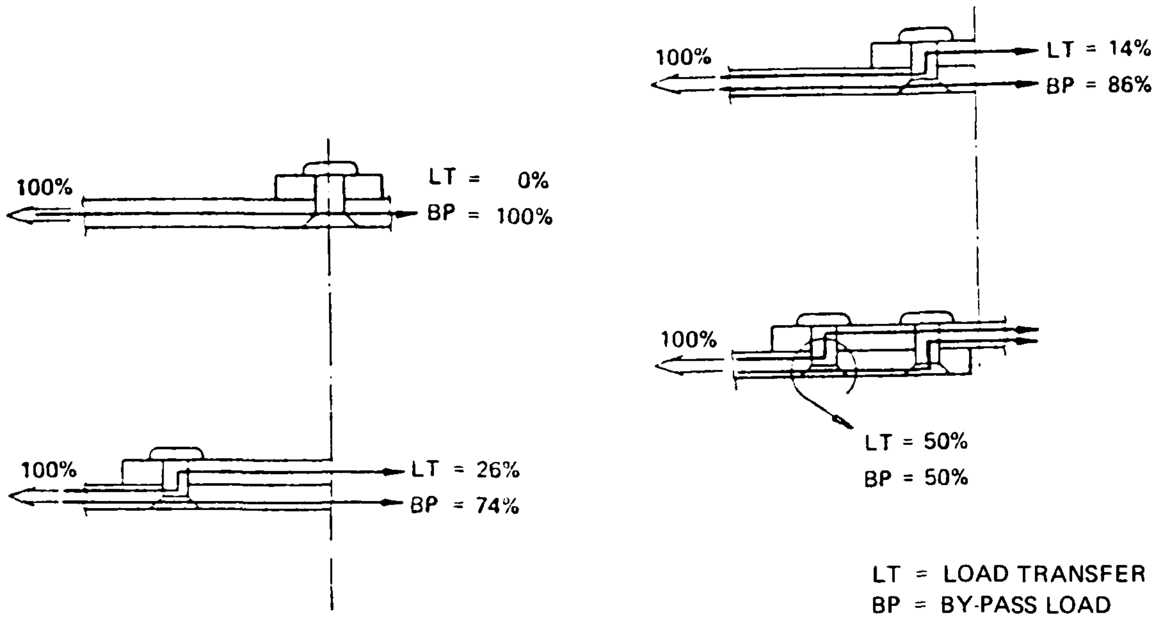


FIG. 22: THE EFFECT ON THE AMOUNTS OF LOAD TRANSFERS AS A RESULT OF GEOMETRICAL CHANGES (REF. 110)

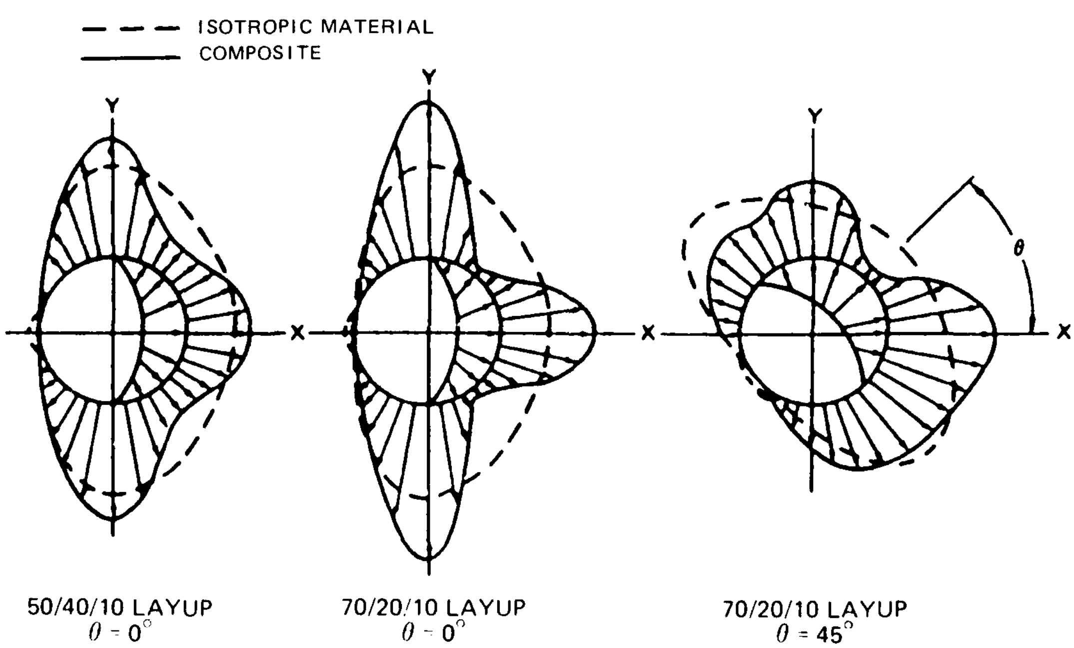


FIG. 23: THE EFFECTS ON CIRCUMFERENTIAL STRESS DISTRIBUTIONS AS A RESULT OF SHIFTING FASTENER LOAD DIRECTION (REF. 56)

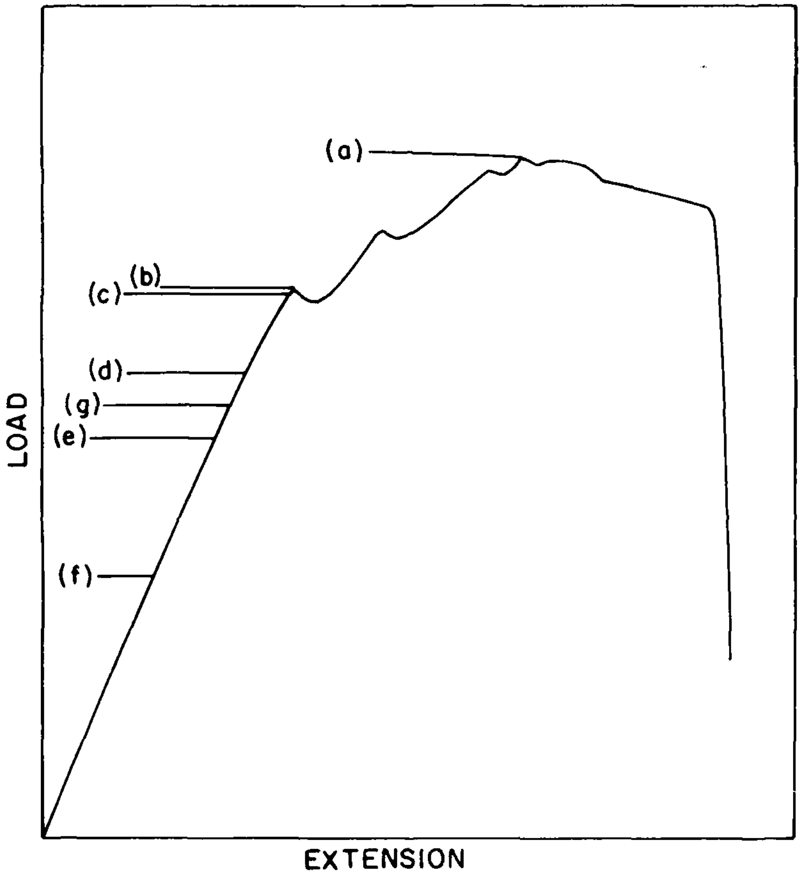


FIG. 24: TYPICAL LOAD/EXTENSION PLOT (REF. 97)

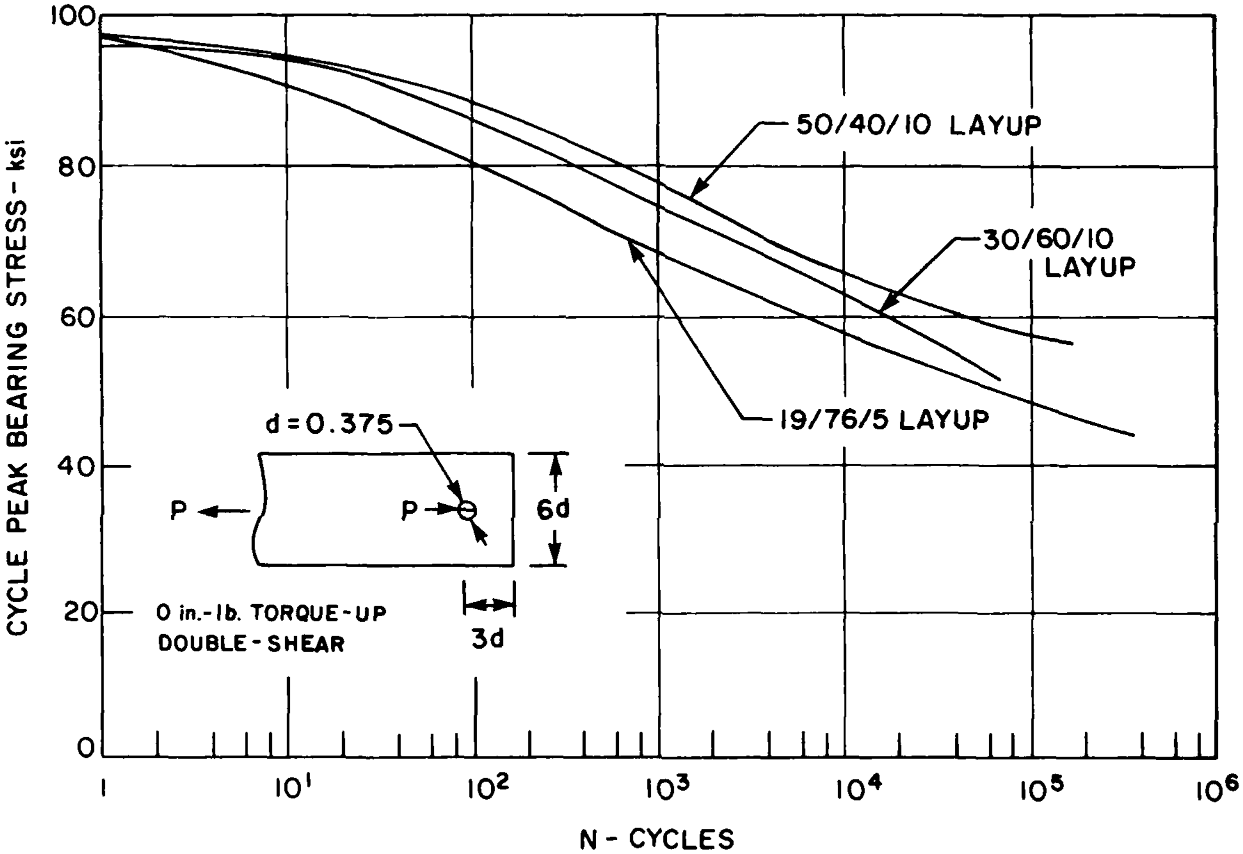


FIG. 25: COMPARISON OF R = -1.0 JOINT FATIGUE LIFE TRENDS (REF. 56)

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NOTES

13

DESCRIPTORS (KEY WORDS)/MOTS-CLÉS

1. Composite materials

14

SUMMARY/SOMMAIRE

This report presents a literature review of the state-of-the-art analytical and experimental methodologies adopted in the aerospace industry for the design of mechanically fastened joints in composite structures. Results and conclusions obtained from the published literature relating to the effects of critical parameters, which include composite material system, fastener configuration and joint geometry, on the mechanical behaviour and failure modes of composite mechanically fastened joints are discussed. Further research required to improve the design of composite mechanically fastened joints is identified as a result of this review.

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